A TEXT-BOOK OF

ENGINEERING DRAWING AND DESIGN.

PART II.—MACHINE AND ENGINE DRAWING AND DESIGN.

A TEXT-BOOK OF

ENGINEERING DRAWING AND DESIGN:

INCLUDING

Practical Geometry, Plane and Solid, and Machine and Engine Drawing and Design.

BY

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With Numerous Illustrations, Examples and Test Questions, specially intended for the Use of Students of Technical Schools and Colleges.

PART II.—MACHINE AND ENGINE DRAWING AND DESIGN.

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PREFACE TO THE FIRST EDITION.

THIS book is intended for the use of Engineering students in schools and colleges, and as a text-book for examinations in which a knowledge of Practical Geometry and Machine Drawing is required.

The chief reason which has led to its preparation is that during the time I was engaged in teaching on the Engineering side of Dulwich College, and had charge of the classes in Geometrical and Mechanical Drawing, I found it impossible to obtain a book wherein the problems, or examples, were not accompanied by diagrams which the student could easily *copy*, without in the least knowing to what they referred. In Plane and Solid Geometry there was a lack of properly graduated questions, and such important parts of the subject as problems on loci, the construction of the useful plane curves and their practical application to cams and wheel-teeth, the interpenetration and development of simple solids, and isometric projection were only to be found in advanced books—although really more suited for elementary students than the troublesome problems on

"points, lines, and planes" which usually precede them. In Machine Drawing and Design, again, the deficiency was still greater, there being no choice between sheets of Diagrams and Text-Books which, although they produce admirable copyists, are utterly devoid of any utility as regards *education* in design. The well-known treatise by Prof. Unwin is, of course, most excellent as an aid to design, but it does not profess to teach drawing, and is certainly not intended for elementary students.

Under these conditions, I found myself obliged to arrange questions so graduated in regard to sequence and difficulty as to be really helpful in "teaching" the subject, by bringing out important principles, by making clear mathematical relations, and by requiring the application of real thought and the knowledge gained in other classes and subjects. The setting of the questions was always preceded by a lesson in which, for geometrical drawing, typical problems were worked upon the blackboard and explained; and for machine drawing, the parts concerned were drawn separately or together, or illustrated by models, and the relations of, and reasons for, the shape and size of the different parts made clear. The book, then, has grown out of my own felt wants, and the effort to supply them by the questions and lecture-notes mentioned.

In Part I., I have included chapters on those parts of Practical Geometry already referred to as usually taken later, because I believe them to be essential to a good elementary course of Practical Geometry, and admirably suited for the ordinary engineering or technical student commencing without previous knowledge, and desiring

to go on to an intelligent study of Machine Drawing and Design. I have also included some special cases of intersection, such as occur in metal plate work and in the drawing of some engine parts, in order that students may have no excuse for putting in the necessary curves by "guess work." Thus, in preparing Part I., I have steadily kept in view the work of Part II., from the conviction that Plane and Solid Geometry should always precede Machine Drawing, just as Arithmetic precedes Algebra.

In Part II., I have avoided dimensioning the illustrations except in rare cases, and have endeavoured to build up the subjects so that all primary and common parts are first explained and understood, such explanations not being repeated when the parts occur in connection with larger or complete designs. Such a method will, I am sure, commend itself to all true teachers. A student ought not to be told the sizes of bolts and nuts, or the diameter of flanges, or the details of stuffing boxes in drawing an engine cylinder, any more than we should expect to have to prove to him the truth of the triangle of forces, at each step in the graphical determination of the stresses in a roof-truss. But such an arrangement obviously requires that the examples be worked through in the order given, and especially is this so in the Sections on Engine Design, the examples in which have been intentionally arranged to show the interdependence of the different parts. I have throughout endeavoured to give the reasons for all features of the designs; when these are purely empirical or for workshop convenience, this is stated. My object will have been attained if I have made it impossible for a student to

draw any part *without* having an intelligent reason for all he does.

It has certainly been my desire to make the book suitable for beginners, believing that the sizes and arrangements of simple common parts cannot be acquired too early, and I have therefore endeavoured to teach the principle of "drawing" as well as of "design." I have found some difficulty in deciding what terms and definitions to employ in order to make the book acceptable to the ordinary student and teacher, and yet free from unscientific and ambiguous expressions. Workshop terms are not fully satisfactory, for they vary with different localities, while scientific terms are often misused. I have restricted the word "compression" to represent a strain or change of form, and in other ways have adhered to the following notation:—

Stress.		Strain.
Pressure or Thrust	produces	Compression.
Tension		Extension.
Shearing Stress		Shear.

My thanks are especially due to Mr. F. W. Sanderson, M.A., Head-Master of Oundle Grammar Schools, who, during my association with him at Dulwich College, gave me much assistance in preparing the Examples, and who has been good enough to read the proof-sheets and to make many valuable suggestions, and to Professor Goodman, M. I. Mech. E., Assoc. M. Inst. C. E. of the Yorkshire College, Leeds, for much helpful counsel as to the arrangement of Part. II. I am indebted also to Mr. J. H. Wicksteed, M.I.C.E., M.I.Mech.E., for useful suggestions and drawings, and to Messrs. Allan & Co., Lambeth; Messrs.

Marshall & Sons, Gainsborough; Mr. W. Allchin, Globe Works, Northampton; The Kirkstall Forge Co.; The Globe Engineering Co.; Messrs. Schaffer & Budenberg of Manchester; and the Atkinson Gas Engine Co., for kindly supplying drawings for insertion in the text. Finally, I have to acknowledge the assistance received from the works of Professors Unwin and Ripper, and Mr. Henry Angel.

I shall be grateful to teachers and others who may use the book for information as to any errors which may have been overlooked.

SIDNEY H. WELLS.

BATTERSEA POLYTECHNIC, S.W.,
September, 1893.

NOTE TO THE FOURTH EDITION.

THE Publishers note with satisfaction the popular esteem in which this work is held. Three Editions have been exhausted, and the opportunity has been taken of this re-issue to add as an Appendix to Part I. a series of Questions set in the Board of Education Examinations, which will be found useful in helping Students to test their knowledge.

January, 1905.

CONTENTS OF PART II.

NOTE ON EXAMPLES.

S. and A. E. = Science and Art Department, Elementary.
S. and A. A. = ,, ,, Advanced.
S. and A. H. = ,, ,, Honours.
V. U. O. = Victoria University, Ordinary.
V. U. H. = ,, ,, Honours.

PART II

MACHINE AND ENGINE DRAWING AND DESIGN.

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SECTION XIII.

INTRODUCTION.

ENGINEERING drawing in its generally accepted sense really consists of two distinct parts, the first being the mechanical act of merely drawing an object in accordance with certain recognised laws of projection; the second, that of designing and deciding the proportions and form of the object by considering the strength of the materials of which it is made, the forces which will act upon it, and the work which it will be called upon to do.

It is scarcely possible to become a good designer without possessing some powers of draughtsmanship, but it is quite easy to be an excellent draughtsman without knowing anything of the principles of design and construction. For instance, a student who has acquired a knowledge of practical geometry, and who, therefore, knows the meaning and uses of "plans," "elevations," "sections," "interpenetrations," and "developments," could, if given sufficient sketches and measurements, make complete drawings of the most complicated machine, and yet be quite ignorant of the manner in which the shape and proportions of the machine had been arrived at.

But such a condition of mere mechanical working is above all things to be avoided, and the objects of the following pages is to show in an elementary manner, the general principles of theory and practice which guide the design and construction of engineering machines and structures. In order that they may be intelligently approached, the student should endeavour to understand the following remarks on drawing and design.

(1) **Drawings.**—Engineering drawings may be divided into two classes :—I. "*Working drawings*," which show all the details

of separate parts, so that the parts can be made from them. II. "*General-arrangement drawings*," showing the complete machine, and the relation and position of the different parts. Usually the designer first makes a rough general-arrangement drawing, in order to obtain the leading sizes of the different parts, after which a complete working drawing of each part is made and sent into the workshops, in order that the part may be at once proceeded with; and, finally, a finished general-arrangement drawing is completed.

(2) **Scale of Drawings.**—In working drawings the scale should be as large as possible—3″, 4½″, or 6″ to 1′ being usual proportions. All small parts should be drawn full size. General-arrangement drawings are made to smaller scales, such as ½″, ¾″, 1″, 1½″, or 3″ to 1′. The scale adopted depends to some extent upon the size of the drawing papers, the usual sizes being:—Double Elephant, 40″ × 27″; Atlas, 34″ × 26″; Imperial, 30″ × 22″; Half-Imperial, 22″ × 15″. In making scale drawings, lengths should be marked off by placing the edge of the scale along the line, and marking the line at the required lengths, *not* by first measuring the length on the scale with a pair of dividers, and then transferring to the drawing.

Always write the scale of a drawing prominently upon it (see p. 14).

(3) **Number of Views in Drawings.**—The chief desirable feature in all engineering drawings is *clearness*, and the number of views and sections must be enough to obtain this; anything further is a waste of time. The student must, therefore, decide the number of views necessary as a first step, since that obviously affects the scale of the drawing. Two views, at least, are invariably required; either one elevation and a plan, or one front and one end elevation; but, in the majority of cases, it is necessary to show three views—two elevations and one plan. With complicated parts or general arrangements, a larger number of views may be necessary in order to show all the details. When an object is symmetrical about a centre line, it is sometimes sufficient to show one-half of the view; also, when one part only of an object requires an additional view, it is only necessary to project that part, and not the whole object. For example, the construction of the inclined bearing in Fig. 157 is shown much better and more quickly by the plan of the cover drawn above the elevation, than by a geometrically correct plan, which would show the cover and nuts foreshortened, owing to their inclination.

(4) **Sections.**—It is only necessary to draw parts in section where their construction cannot be clearly shown without doing

so. In many cases it is sufficient to show the parts by dotted lines, or it may be better to show a part only in section and the remainder by dotted lines. When an object is symmetrical about a centre line, it is usual to show one-half only in section, as seen in Figs. 154*b*, 164. The section may end exactly at the centre line, as in Fig. 154*b*, or it may extend beyond it, as in Fig. 153. The end of the section part is shown by black lines.

Objects which are completely solid, such as bolts, rods, spindles, and shafts, are not shown in section, although, strictly speaking, they are cut by the section plane as much as the other parts. Therefore, in drawing the section of a stuffing box, as in Fig. 163, or of a valve, as in Fig. 172, the rod, spindle, and the studs are not sectioned. In the same way the arms of a wheel (Fig. 174), the guides of a valve (Figs. 173, 175), and the nuts of bolts and screws are not shown in section. Nothing would be gained by sectioning such parts, and a drawing is much clearer by leaving them unsectioned.

(5) **Dotted Lines.**—The student will have seen from preceding examples that dotted lines are used to show parts of an object that are not really seen from the point at which the object is viewed. In engineering drawings it is often necessary to make a free use of dotted lines, but they should always be avoided when they do not really add to the clearness of a drawing. In such sectional views, as of the stuffing box in Fig. 163*b*, and of the shaft-coupling in Fig. 137, it would be strictly accurate to show by dotted lines the edges of the flanges on the back of the object, but as clearness is not generally gained by so doing, they are better omitted.

(6) **Order of Drawing Centre Lines.** — It seems to be a constant difficulty with a student to know where to begin a drawing. Few things look more careless than the different views of a drawing crowded together or near one edge of the paper, when there is plenty of room for a good space between the views and for an equal border all round. The order of drawing should be as follows :—

I. Decide the number of views and the scale.

II. Find approximately the space each view will take up and the position of the chief centre lines.

III. Pencil in the chief centre lines of the different views in such a position as to give equal space between the views and an equal border all round when the drawing is finished.

IV. Draw the leading part in all the views, and then add the remaining parts in the order of their relation to the leading part and to each other. In each case show one part in all the views before commencing another part, so that the different views are

drawn together, as if one view is completed before another is commenced it is impossible to correctly follow the relation of the parts, and the drawing takes very much longer to finish. It is often convenient to draw in first those parts to be shown by full lines, as it is then easier to see that other parts which come under or behind them must be shown by dotted lines.

All such details, as bolts and nuts, keys and other small and repeated parts, should be left until the size and position of the leading parts have been finally fixed and drawn, otherwise an alteration in a leading part means a rubbing out of all the details. When the object contains a number of similar parts such as bolts and nuts, the similar circles and lines of each should be drawn in together, and not by finishing each one separately. As far as possible, the student should aim at not having to take a dimension for the same part from his scale or rule more than once.

In many of the following examples an effort is made to show the order in which the different parts should be drawn, for it is quite impossible to design properly unless the relation of the different parts to each other is fully and clearly recognised.

V. In inking-in, always finish all circles and arcs of circles first, and then the straight lines. It is not generally the quicker way to ink-in one view completely before commencing another, but to work all together. In inking-in the straight lines it is best to put all the lines which go in one direction first, and afterwards all those in another direction. *Centre lines should be inked-in as full red lines.*

(7) **Dimensioning a Drawing.**—All the parts of working drawings should be fully dimensioned the actual size of the part, blue ink being used for the dimension lines, *which should not be dotted*, and black ink for the figures and arrows. The student should aim at showing the dimensions as clearly as possible, and should group the dimensions for each part together, so that it is not necessary to hunt all over the drawing in order to find them. By referring to the dimensions on Figs. 159, 162, it will be seen that they are arranged either on the drawing itself, or slightly removed from it, whichever most ensures neatness and clearness. Also notice that vertical dimensions should be arranged to read from the bottom edge of the drawing to the top edge, the eyes being turned to the left hand. It is not necessary to dimension general-arrangement drawings with the exception, perhaps, of the chief sizes.

(8) **Colouring and Shading.**—Different colours are used to show the metals of which parts are made, the usual arrangement being as follows:—

Metal.	*Colour.*	
Wrought steel, ...	Bluish-purple	Prussian blue and Crimson-lake.
Cast steel, ...	Reddish-purple	
Wrought iron, ...	Prussian blue.	
Cast iron, ...	Paynes grey or Neutral tint.	
Brass and gun-metal, . .	Chrome-yellow.	
Copper, ...	Crimson-lake.	
Wood, ...	Burnt sienna, streaked with Vandyke brown.	

In highly coloured drawings it is usual to show holes by a light wash of black.

In drawings where no parts are shown in section, it may be necessary to use a little colour to denote the metals, and this is best done by a narrow band of colour inside the outer lines. Generally speaking, all colours should be used in as light washes as possible; but in drawings where parts both in section and not in section are coloured, it is usual to indicate the sectioned parts by using the colour of a darker tone. All sectioned parts should be coloured, either by washing all over or by hatching the parts by broad diagonal lines of colour. The latter is usual for drawings of larger scale, as large patches of colour are to be avoided. Drawings which are not coloured should have their sectioned parts shown by ink lines drawn diagonally, as in the following figures (see p. 119).

The shading of inclined or circular parts is affected either by "line shading" or "colour shading," in the former a flat incline is shown by drawing a number of equally spaced parallel lines (Fig. 102*a*), and a circular part by lines which are drawn closer

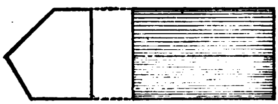


Fig. 102*a*.

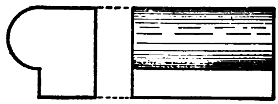


Fig. 102*b*.

together as they approach the outside lines of the diameter (Fig. 102*b*). Colour shading consists of graduating the tone of the colour from light to dark to produce the same effect, this is best done by a wash of black or Indian ink, and afterwards washing over with a light uniform tint of the required colour. General arrangement drawings are frequently highly coloured and shaded.

The student should occasionally practise these methods of shading, in order that he may be able to use them if required. In drawing offices the practice as to colouring and shading varies

considerably, and generally depends upon the amount of time at the disposal of the draughtsman. The same remark applies to the title of a drawing; this should always be clearly written or printed in a conspicuous place, but the question of whether it shall be plainly or elaborately done also depends upon special circumstances.

(9) **Design.**—The first principle of design is to arrange the form of a machine or structure according to the work it is required to do, and the second to proportion its different parts in accordance with the known forces it will have to resist, and the resistance of the material of which it is made. The former of these is probably a matter of personal intuition, and the latter is considered as shown more fully in later pages. But in addition to these, most engineering machines have to be designed by considerations of practical workshop conveniences and possibilities, of the cost of construction, of convenience in repairing, and of simplicity and symmetry of form. The full importance of these points cannot be recognised except by those possessed of practical workshop experience; but if the student will carefully consider the following remarks, he will be materially assisted in working through the remaining sections of this book, and in approaching the problem in an intelligent manner.

(10) **Cost of Construction.**—First of all it should be understood that the design generally most approved by the engineer is that which is the most easily and cheaply constructed, and which offers the best facilities for repair. There are many arrangements of forgings and castings which may appear at first sight very suitable, but which are more difficult to forge or cast than some other design not apparently so convenient and neat, and in the same way a part may be designed in order to permit of machining in a certain way, although by so doing it may even be wanting in good proportions and form. It is well known that "turning" is the cheapest kind of machining, and, as a result, there are a large number of parts which are designed with the chief object of getting in as much turning and as little of other machining as possible, although other considerations would seem to suggest quite another form of construction. Designers also aim at keeping the amount of machining as small as possible, for which object the parts of castings which require facing for fitting together or to some other piece, are made with projecting bosses, lugs, or strips which allow of machining without touching the main body of the casting. It is only by considerations such as these that many of the designs in engines and machines can be explained.

(11) **Proportions of Parts.**—Then, again, with regard to the

proportions of different parts. It may often be possible to determine exactly the forces which will act upon any given piece of a machine, and then by knowing the safe working stress to which it may be subjected to arrange the proportions accordingly. The strength and ordinary working stresses of different materials are given in the following pages, but if any special material is used the value of its resistance and its safe working stress should be determined before proceeding to the design.

But there are undoubtedly many parts of machines where it is practically impossible to find the stresses which act upon them when working, as, for instance, in lathes and other shop tools, or in spinning and weaving machinery, and, similarly, there are other parts, such as brackets and pedestals for shaft bearings, in which, although the forces acting upon them may be measurable, yet their resistance to these forces is far too difficult for ordinary calculation owing to their complex form.

Such parts have then to be proportioned in accordance with practical experience, by knowing what has been allowed in similar cases before, and also by such assistance as can be gained by a general consideration of symmetrical proportion and convenient arrangement. It is probable that all such parts are really abnormally strong for the work they have to do, but it must also be remembered that stiffness and solidity may be absolutely necessary, to obtain which much metal is required.

(12) **Castings.**—And, still further, it should be noticed that castings are invariably much heavier and stronger than forgings, on account of practical foundry difficulties, which soon reach a limit of possible thinness. The metal must also be more equally spread over the different parts on account of the stresses produced when cooling, and in order to ensure an equal cooling of the whole mass. *Notice also that the corners of castings are always well rounded.*

These are, after all, but a few of the many points which have to be considered in engineering design, but so far as they go, they should be grasped by the student. He will then, perhaps, be better able to understand why so many machines, which come under his notice, appear so abnormally strong, heavy, and ugly, and to see how impossible it is to lay down hard and fast rules for the proportions of each part, on account of the different degrees of importance which may be given to the various points which have been mentioned. But he should also recognise the necessity of approaching the question of design in as scientific and thoughtful a manner as possible, of bringing to bear upon it all the knowledge he possesses of the strength of materials and their most efficient use, and of seeing that he should not neces-

sarily be tied by any particular proportion or arrangement when several ways are open, unless his choice be directed by sound and sensible considerations. It would be a most invaluable result if every student of machine design would take care never to draw any part, no matter how unimportant, without having fully considered the whole "why and wherefore" of the part, and being prepared to sensibly justify his design if challenged.

SECTION XIV.

STRENGTH OF MATERIALS.

THE scope of this work is not sufficient to require any detailed or advanced treatment of the strength and properties of engineering materials. But the obvious fact that the strength of machines and structures must bear some relation to the strength of the materials of which they are made, necessitates at least a clear understanding of the different forces which act upon materials, their classification and recognition, and the power possessed by the materials to resist them.

(13) **Stresses in Materials.**—Generally speaking, all materials of construction are called upon to resist stresses caused by their own mass as well as those produced by external forces. In the case of heavy structures, such as bridges or arches, these stresses may be very great, and in some instances may even exceed the external forces, hence they cannot then be neglected, but in most machines the masses of the parts themselves need not be considered, except when their momentum produces stress; and we only have to deal with the external forces acting upon the material, caused generally by the work the machine does, and capable of a sufficiently exact measurement.

The forces which act upon materials may be classified as follows:—

(*a*) *Tensile forces.*—Those which tend to separate the particles of a body from each other by direct pulling.

EXAMPLES.—Forces in ropes, chains, belts, bolts in flanges and covers, connecting-rods during inward stroke, tie bars of roofs and bridges.

(*b*) *Compressive forces.*—Those which tend to press together the particles of a body by direct pushing.

EXAMPLES.—Forces in supporting columns, rams of pumps and presses, connecting-rods during outward stroke, struts in bridges and roofs, jibs in cranes.

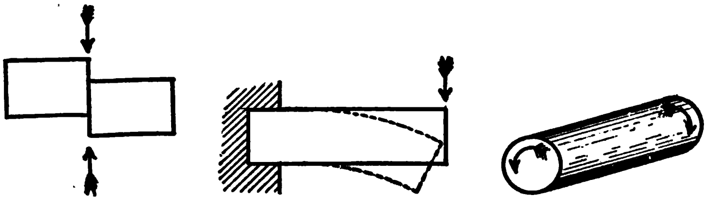
(*c*) *Shearing forces.*—Those which tend to cause the particles of a body to move one over the other by direct sliding.

a *b*

EXAMPLES.—Forces on rivets in riveted joints, bolts in shaft couplings, pins and cotters, shafts transmitting power.

(*d*) *Bending or Transverse forces.*—Those which tend to make a body assume a position at an angle to its former position.

EXAMPLES.—Forces acting on beams, hooks of cranes.



c *d* *e*

(*e*) *Torsional forces.* — Those which tend to separate the particles of a body by direct twisting. When a body is twisted apart, its particles separate, due to a shearing force.

EXAMPLES.—Forces in shafts transmitting power.

(14) **Measurement of Forces.**—It is clear that the effect produced by tensile, compressive, or shearing forces depends upon the magnitude of the forces, and the area across which they act, and is completely measured when we know, for instance, that the force is equal to, say, 5,000 lbs. or 5 tons on every square inch. But it is equally clear that the effect of a force producing, bending, or twisting depends not only upon the magnitude of the force, but also upon the distance at which it acts from the point, about which the bending or twisting is taking place. Hence, we must consider the combined effect of the force and the distance, and following the laws of mechanics we thus speak of bending moments (B. M.), and twisting moments (T. M.) Bending and twisting moments are due to external causes, and produce in the

material on which they act, tensile and compressive stresses and shearing stresses.

(15) **Working Stress.**—It is not so important for the designer to remember the ultimate resistance or "breaking strength" of the material he uses, as to know the working stress or "safe load" to which the material may be subjected in actual construction. This safe load evidently depends upon the kind of work the material has to do, and the nature of the stress it has to withstand. For instance, the safe load may be higher in such cases as columns of buildings, where the load remaining practically still and unchanged is termed a "*dead load*," than in the case of railway bridges, where the load is constantly changing and is known as a "*live load*," while again parts of machines, such as the piston and connecting-rods of engines must have a still lower working load in consequence of the stresses being alternately tensile and compressive.

The following tables give fair average values of the breaking strength and safe working stresses under the conditions referred to for the common materials of construction, which will be adhered to throughout this work:—

TABLE I.—Breaking Strength and Weights.

(For Short Pieces only in Thrust.)

Metal.	Tension per Square Inch.		Thrust per Square Inch.		Shearing Stress per Square Inch.		Weight.	
	Tons.	Lbs.	Tons.	Lbs.	Tons.	Lbs.	Lbs. per cu. ft.	Lbs. per cu. in.
Cast iron, . .	8	18,000	45	100,000	6	13,000	468	0·271
Wrought iron, Bars,	23	51,500	23	51,500	20	45,000	490	0·283
Wrought iron, Plates ‖	21	47,000	...	...	17	38,000	490	0·283
Wrought iron, " +	18	40,000	...	...	...	...	490	0·283
Mild steel, . .	30	67,000	30	67,000	24	54,000	480	0·278
Gun-metal, = Cu + tin, .	13	30,000	...	...	...	...	546	0·316
Brasses, = Cu + zinc, .	11	24,500	...	...	...	...	513	0·300
Copper, . .	13	30,000	...	...	...	...	555	0·321

TABLE II.—ORDINARY WORKING STRESSES IN POUNDS PER SQUARE INCH (*Unwin*).

Metal.	Tension and Thrust.			Shearing Stress.			Torsion.		
	Dead Load.	Live Load.	Live Load changing Stresses.	Dead Load.	Live Load.	Live Load changing Stresses.	Dead Load.	Live Load.	Live Load changing Stresses.
Cast iron, . . .	4,200	2,800	1,400	...	2,300	...	2,100	1,400	700
Wrought iron, Bars,	15,000	10,000	5,000	12,000	8,000	4,000	6,000	4,000	2,000
Wrought iron, Plates ‖	15,000	10,000	5,000	...	...	...	...	...	...
Wrought iron, ,, +	12,000	8,000	4,000	10,000	6,600	3,300	...	...	...
Mild steel, . . .	20,000	13,200	6,600	16,000	10,600	5,300	8,000	5,400	2,700

N.B.—The live load as above is two-thirds the dead load, and the "live load with changing stress" is half the live load.

It is convenient to remember that the shearing strength of wrought iron and steel is approximately four-fifths of the tensile strength; and that the tensile and compressive strengths are usually taken as equal.

(16) **Stress proportioned to weakest Section.**—In calculating the strength of separate portions of a machine or structure, care must be taken to allow for all the different stresses it may be subject to, and to make it sufficiently strong in its weakest direction. For instance, if a piece of cast iron be alternately acted upon by equal tensile and compressive forces, its section must be calculated as a case of tension, because cast iron offers so much less resistance to tensile than to compressive forces. In like manner, with a piece of unequal section subject to uniform stress, its strength must be calculated on its smallest section, since that is obviously where it is most likely to yield first. It is for this reason that in all screwed bolts or spindles *the section at the bottom of the thread* must be taken as the effective area resisting breaking.

(17) **Units of Measurement.**—The unit of area adopted in the following pages will be "*the square inch*," and the unit of force "*the pound*," or "*the ton*," hence stresses will be expressed either as "*lbs. per sq. inch*," or as "*tons per sq. inch*."

The following expressions will also be adhered to as representing the *safe working stress* per sq. inch:—

f_t for tensile.
f_c for compressive.
f_s for shearing stresses.

As a general rule, the letter *d* will be used to represent diameter; *d* meaning diameter of rod or rivet, d_s diameter of shaft or spindle; and so on.

SECTION XV.

BOLTS AND NUTS—SCREW THREADS.

(18) Bolts, Studs, and Screws.—Bolts and nuts are the most common form of temporary fastenings employed in engineering construction. In general machine practice they are made as in Fig. 103, where the bolt head and nut are of hexagonal shape, either left "*black*" as forged, or "*finished*" by machinery, according to the quality of the work on which they are used.

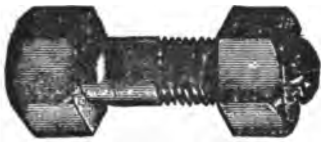


Fig. 103.

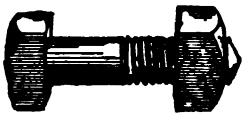


Fig. 104.

In rough work, such as bridges, roofs, and similar structures, a much cheaper make is used, where the head and nut are square in shape, as in Fig. 104.

For work where only the nut is seen, a square head bolt, or a black hexagon bolt with a finished hexagon nut, is used.

It is not always possible, as will be more clearly seen later,

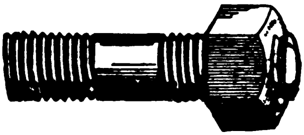


Fig. 105*a*.

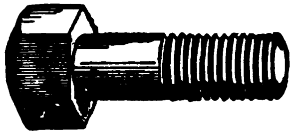


Fig. 105*b*.

to use this form of bolt and nut. There are two other forms in common use, one of which consists of a piece of rod screwed at both ends and used with one nut, as in Fig. 105*a*, called a "*stud*," the other, differing but slightly from a bolt without a nut, called a "*screw*" (Fig. 105*b*). That part of the body of a bolt, stud, or screw, left unscrewed, is called the "*plain part*."

(19) Table of Standard Bolts and Nuts.—The adoption in this country of a universal standard of screw threads, and of sizes for bolts and nuts, is due to the late Sir Joseph Whitworth, and is known as the "*Whitworth Standard.*" All ordinary bolts and nuts are made to these sizes, the screw thread adopted being known as the "*Whitworth*" or "*V*" thread.

Diameter.—By the diameter of a bolt or nut is meant the diameter of the plain part, or the diameter outside the threads. It is, of course, the diameter to which a spindle is turned before a thread is cut upon it. By a 1″ nut is meant a nut for an inch bolt.

Nuts.—The thickness of the nuts in all ordinary work is, in every case, equal to the diameter of the bolts.

TABLE III.—Sizes of Whitworth Standard, Bolts, Nuts, and Screw Threads.

Diameter of Bolt.	Head and Nut across Flats.	Thickness of Head.	Number of Threads per Inch.	Diameter at Bottom of Threads.	Area at Bottom of Threads.	Weight in Pounds, including Head.	
						Per Inch Length of Bolt.	Nut and Part of Bolt in Nut.
					sq. inches.		
$\frac{1}{2}''$	$\frac{7}{8}''$	$\frac{1}{2}''$	12	0·393″	0·121	0·057	0·110
$\frac{5}{8}''$	$1\frac{1}{8}''$	$\frac{5}{8}''$	11	0·509″	0·203	0·087	0·172
$\frac{3}{4}''$	$1\frac{5}{16}''$	$\frac{11}{16}''$	10	0·622″	0·303	0·124	0·324
$\frac{7}{8}''$	$1\frac{1}{2}''$	$\frac{3}{4}''$	9	0·733″	0·422	0·170	0·494
1″	$1\frac{11}{16}''$	$\frac{7}{8}''$	8	0·840″	0·554	0·220	0·681
$1\frac{1}{8}''$	$1\frac{7}{8}''$	$\frac{15}{16}''$	7	0·942″	0·697	0·280	0·873
$1\frac{1}{4}''$	$2\frac{1}{16}''$	1″	7	1·067″	0·894	0·340	1·188
$1\frac{3}{8}''$	$2\frac{1}{4}''$	$1\frac{1}{8}''$	6	1·161″	1·060	...	...
$1\frac{1}{2}''$	$2\frac{7}{16}''$	$1\frac{1}{4}''$	6	1·286″	1·300	0·490	2·088
$1\frac{5}{8}''$	$2\frac{9}{16}''$	$1\frac{3}{8}''$	5	1·369″	1·472	...	...
$1\frac{3}{4}''$	$2\frac{3}{4}''$	$1\frac{7}{16}''$	5	1·494″	1·753	0·680	2·954
$1\frac{7}{8}''$	3″	$1\frac{1}{2}''$	$4\frac{1}{2}$	1·590″	1·987	...	...
2″	$3\frac{1}{8}''$	$1\frac{5}{8}''$	$4\frac{1}{2}$	1·715″	2·311	0·880	4·387
$2\frac{1}{4}''$	$3\frac{9}{16}''$	$1\frac{3}{4}''$	4	1·930″	2·925	...	...
$2\frac{1}{2}''$	$3\frac{7}{8}''$	$1\frac{7}{8}''$	4	2·180″	3·732	1·37	8·50
$2\frac{3}{4}''$	$4\frac{3}{16}''$	2″	$3\frac{1}{2}$	2·384″	4·465	...	...
3″	$4\frac{1}{2}''$	$2\frac{1}{4}''$	$3\frac{1}{2}$	2·634″	5·450	1·98	14·14

N.B.—The following rule is a useful one to remember:—Size across flats equal $1\frac{1}{2}$ D + $\frac{1}{8}$ in inches.

Square nuts and bolt heads are made the same size across flats as hexagon nuts and heads.

(20) **Pitch.**—By the pitch (p) of a screw thread is meant the distance between the centres of two threads next to one another. A screw with eight threads per inch of length is, therefore, of $\frac{1}{8}''$ pitch, and a nut upon it would advance $\frac{1}{8}''$ for every revolution.

The same word is used for the distance between the centres of

bolt or rivet holes, so that we speak of bolts of 5″ pitch, or rivets of $2\frac{1}{4}$″ pitch.

(21) **Whitworth or V Thread.**—A section of this thread is shown in Fig. 106. The angle between the sides of the thread is 55°, and each thread is rounded off at the top and bottom one-sixth of the total depth. This form gives great strength, combined with great frictional grip, when an inside and outside thread are screwed together, and there is thus considerable resistance offered to nuts or bolts working loose. These properties make it especially useful for fastening purposes, but lessen its efficiency for purposes where much movement is necessary, as for adjusting pieces or for transmitting work in machines.

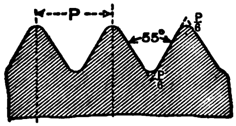
(22) **Square Thread Screw.**—The next most common form of screw is known as the square thread, owing to the sides of the thread being perpendicular to the axis of the spindle. Its form in section is shown in Fig. 107. The width of the thread is equal to the space between the threads, and is, therefore, equal to half the pitch. The depth of the thread is $\frac{19}{40}\,p$ when accurately made, but in drawing the thread it is sufficiently near to make the depth equal to half the pitch. The pitch of a square thread is usually made twice the pitch of a V thread of the same diameter, it has, therefore, only half the number of threads per inch of length. There is very little frictional grip between this thread and nut, and, therefore, less wear than with the V thread; it is consequently well adapted for screws subject to much movement, such as the leading and feeding screws of lathes, and other machines for which it is largely used.

(23) **Knuckle or Rounded Thread Screw.**—A modification of a square thread, known as the rounded or knuckle thread, has the top and bottom of the threads rounded as semicircles, as shown by dotted lines in the right-hand threads of Fig. 108. Its advantages are, that it is less liable to injury from blows, having no square corners to be damaged, and it slips more easily into or out of a portion of a nut when requiring to be thrown in or out of gear, as in the saddle of a screw-cutting lathe.

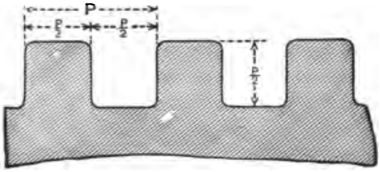
(24) **Buttress Thread Screw.**—This is a combination of the V and square threads, and is shown in Fig. 109, one side being square with the axis, the other inclined at an angle of 45°. It is used largely where the resistance is in one direction only, as in breeches of guns, screw presses, &c.

(25) **Drawing Screw Threads.**—Strictly speaking, a screw thread is a helix, and if properly drawn requires a somewhat lengthy geometrical construction, as already described in Problem lvii. In ordinary workshop drawings this would mean a great

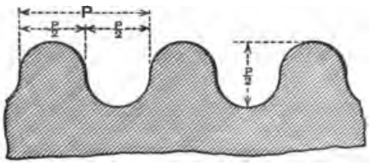
waste of time, and as a consequence the following approximate methods are adopted in drawing offices :—



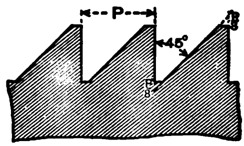
WHITWORTH VEE THREAD.
Fig. 106.



SQUARE THREAD.
Fig. 107.



ROUNDED THREAD
Fig. 108.



BUTTRESS THREAD.
Fig. 109.

V Threads (Fig. 110, *a, b, c*).—(*a*) For sizes up to 1″ or 1¼″ diameter the screw is drawn in by hand as shown; with larger sizes the proper number of threads per inch may be set out, and a 55° set square used. After a little practice it is possible to show threads in this way with good accuracy and considerable ease.

(*b*) Same as (*a*), but adding the cross lines, those joining the bottoms of the threads being darker than those joining the tops.

(*c*) Parallel lines, representing the diameter, are drawn for the whole length of the bolt, and cross lines, at the proper angle and distance apart (½ the pitch), are then drawn as shown, the darker alternate lines not going right across and representing the bottoms of the threads. This is perhaps the neatest method, and will be generally adopted in the examples of this book.

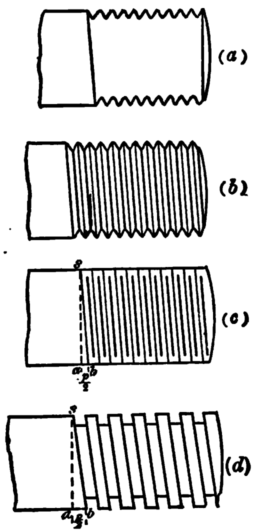


Fig. 110.

To obtain the correct angle of slope of the lines, it is best to set off a distance equal to half the pitch as shown at *a b* (Fig. 110, *c*).

Then the line joining the point *s* to *b* represents one thread, the other lines being drawn parallel.

(26) **Square Threads.**—These are usually represented as shown in Fig. 110, *d*. Start by setting off a distance of $\frac{p}{2}$ to obtain the correct inclination of the lines as in Fig. 110, *c*, and then draw parallel lines a distance apart of $\frac{p}{2}$.

(27) **Right- and Left-Handed Screws.**—Screw threads may be either right-handed or left-handed. The threads in Fig. 110 are all right-handed, and it will be seen that if a nut be turned round on the thread in the same direction as the hands of a watch it will move along the thread away from the starting point. A left-handed thread is exactly the reverse of this, and the lines representing the thread would, therefore, slope from *left to right*, and not as with right-handed threads from *right to left*.

(28) **Gas Threads.**—There are many parts, such as pipes and their connections for gas, water, and steam, which cannot be screwed with such deep threads as the Whitworth standard. The thread adopted is of **V** form but of small pitch, the number of threads per inch of length being as follows:—

Diameter of pipe,	½″	¾″	1″	1½″	2″
No. of threads per inch,	14	14	11	11	11

N.B.—The diameter of iron pipes is measured internally and of brass pipes externally.

EXAMPLES.

(1) Draw correctly in sectional outline and fully dimension four or five threads of (*a*) Whitworth **V** thread, (*b*) square thread, (*c*) buttress thread.

(2) Draw a screw thread upon a cylinder 2″ diameter for a length of 3″ by drawing office method, (*a*) Whitworth **V** thread, (*b*) square thread.

(3) A hollow cylinder 3″ internal diameter, 5″ external diameter, 4″ long, is screwed internally. Draw a longitudinal section showing the thread (*a*) Whitworth **V** thread, (*b*) square thread.

SECTION XVI.

NUTS.

(29) **Hexagonal Nuts** (Fig. 111).—Thickness = diameter (d); size across flats = $1\frac{1}{2}'' \, d + \frac{1}{8}''$.

A hexagonal nut only differs from a plain hollow hexagonal prism in having the corners of one base bevelled or "chamfered" off to give greater finish. Hence the drawing of a nut in different positions is almost identical with the projection of a hexagonal prism, and should present no difficulty after working the problems of pp. 108, 110, 118.

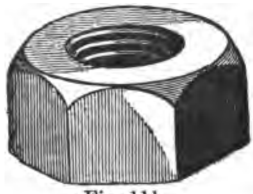


Fig. 111.

The "chamfer" of the corners is usually done at 45° to the axis of the nut, and proceeds until a complete circle is formed on the base.

Many engineers adopt the practice of chamfering nuts on both faces, or if they use nuts only chamfered on one face, they fit them with the chamfer next the work in order to prevent the sharp corners of the nuts cutting the metal against which they bear.

(30) **Drawing of Nuts.**—The number of nuts (or bolt heads) required to be shown on machine drawings is generally very great, and it is necessary to adopt a quick and ready method of drawing them. It is frequently necessary to draw either or both of the views A and B (Fig. 112), without on the same drawing requiring to show the other view C, therefore it is necessary to have a method which will allow of this being done without requiring the separate drawing of the view C.

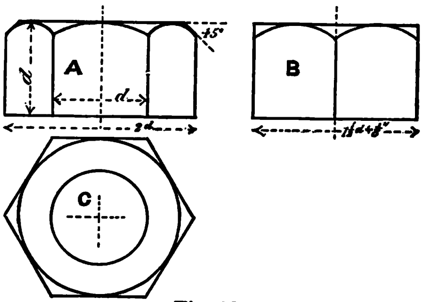


Fig. 112.

The following are the usual methods adopted:—

(a) For showing a nut across the corners (3 faces) as at A, Fig. 112, make the distance across the corners equal to twice the diameter, and the width of the middle face equal to the diameter.

(*b*) For showing a nut across the flats (2 faces) Fig. 112, B, adopt the rule for sizes already given (§ 29).

But when the view C, Fig. 112, must be shown upon a drawing, it is better to draw it before the views A and B, making the circle, round which the hexagon is drawn, equal in diameter to $1\frac{1}{2}\,d + \frac{1}{8}$, and then projecting either the view A or B from it. When this is done the side of A across the corner will not be exactly 2 *d*, especially for nuts above 1″ diameter.

One very important rule, which should be always adhered to, is that when only one of the two views, A or B, is required, the first one should be adopted—that is, *a nut should always be shown across the corners rather than across the flats.* The reason for this is that one seldom wants to know the *greatest* clearance between parts of machines, but the *least* clearance. If a nut is drawn across the flats it does not show the least, but the greatest, and is thus misleading.

Students will find this apparently small point is really one of great importance, especially when designing such parts as flanges, covers, &c., the size of which depends, to a large extent, upon the size of the bolt and nut used.

The sizes of ordinary bolts and nuts need not be marked upon drawings, except the diameter and length. It is a waste of time to dimension the nut or head, or to write on the pitch of the thread except in unusual cases, as bolts and nuts are all made to standard sizes, and the workman only requires to know the diameter and length in order to obtain them from the stores.

The rule as to making the size of a nut across the corners equal to twice the diameter is intended to hold rather for *drawing* a nut than as a safe rule for determining sizes when designing. In such cases it is better to draw the hexagon, knowing the size across the flats equals $1\frac{1}{2}\,d + \frac{1}{8}''$, and measure the size across the corners from it, or to consult Table III. (p. 165).

Fig. 113 shows the drawing office method of showing the chamfer using the rule that, size across corners = 2 *d*. A semicircle is drawn of radius = diameter of bolt, which gives both the height and the limit of the width. The completion of the view is then easily and quickly accomplished. To completely draw a nut it is necessary to draw lines at 45,° as shown in Fig. 112, A, but in most drawings this is omitted.

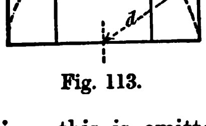


Fig. 113.

(31) **Lock Nuts.**—In pieces subject to rapid movement, which are held together by bolts and nuts, there is a considerable

tendency for the nut to work loose owing to constant vibration, this is prevented by using two nuts, one screwed tightly down upon the other, the top one being termed a *lock nut.* As the duty of this second nut is only to jamb the first and not to take much or any of the stress, it may be much thinner than the usual standard. As a general rule, the thin nut is made equal to half the bolt diameter.

In ordinary practice the nuts are arranged as shown in Fig. 114, the thin nut being the lock-nut. Many authorities, however, say the thin nut should be inside, an arrangement which is not practically as convenient, owing to the fact that ordinary spanners are frequently too thick to admit of fitting on the thin nut when it is so placed. Many engineers use an ordinary nut for a lock-nut, thus having both nuts the same thickness.

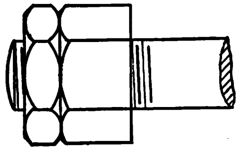


Fig. 114.

Lock-nuts are chamfered on both faces.

(32) **Methods of Locking Nuts.**—There are many other ways of locking nuts. The most common, perhaps, is to drive a taper or split-pin through the bolt just above the nut (Fig. 115*a*), but this scarcely locks the nut, although it prevents it screwing off. A better arrangement is to drive a strong taper-pin with a split end through the nut and bolt when screwed home, the end being then opened out (Fig. 115*b*).

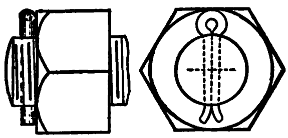


Fig. 115*a*.

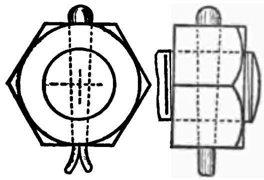


Fig. 115*b*.

Still more certain guards are obtained by using one of the three methods, *a b c* (Fig. 116). In the first (*a*), a set-screw jambs against a part of the nut, turned cylindrical, to form a collar; in the second (*b*), a pin is screwed into the piece against which the nut bears, close up to one face of the nut, diameter of pin not less than ½″; and in the third (*c*), a guard plate is fitted accurately

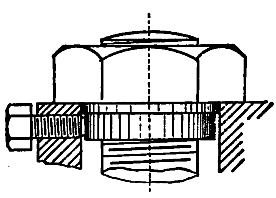


Fig. 116*a*.

against two or more faces of the nut, and screwed down firmly to the body of the piece, thickness of guard plate $\frac{1}{4}\,d$ up to $\frac{3}{4}''$.

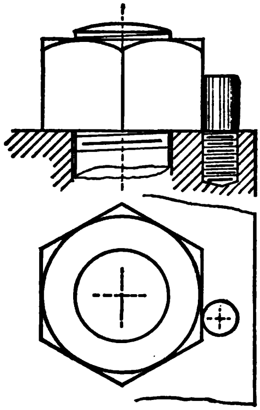


Fig. 116b.

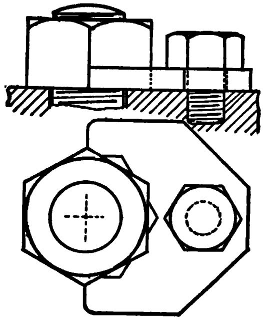


Fig. 116c.

In parts subject to much movement, especially when they are internal, such as piston and valve attachments, great care must be taken to ensure absolute security against unscrewing.

There are numerous other forms of locking arrangements which may be seen largely illustrated in the various engineering journals.

(33) **Washers.**—Washers are thin circular plates made with central holes slightly larger in diameter than the bolts over which they fit, and are frequently used between a nut and the part against which the nut would otherwise bear (Fig. 117). They prevent the nut cutting into the metal, and thus allow of its being screwed more tightly home. They are made to standard sizes, and are used by some makers for almost all purposes where nuts are employed. Their use gives a better finish, and they are also very useful as packing. Spring washers are occasionally used, which, by jambing against the nut, acts as a locking arrangement.

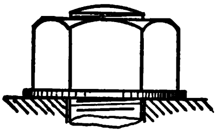


Fig. 117.

Ordinary washers vary in thickness according to diameter, from about $\frac{1}{16}''$ to $\frac{1}{4}''$, or, say, thickness $= 0{\cdot}15\,d$. In diameter they should project beyond the edges of the nut by about $\frac{1}{16}''$ to $\frac{1}{4}''$, according to the size of bolt. It is not necessary to dimension washers, except those of unusual sizes, which

require special making, and it is unnecessary to show them on drawings of their absolute exact size, since they are made in large quantities to certain standard sizes, according to the bolt diameters.

Washers are also necessary when the hole through which the bolt passes is much larger than the bolt, and when the nut would probably only bear upon its corners, or when screwing down nuts against wood or stone. They must then be made of increased thickness—say, equal to $\frac{1}{4}d$.

(34) **Covered Nuts.**—It is sometimes necessary to cover the end of a bolt on which the nut is screwed, to protect the end of the bolt from injury, or to prevent the entrance of dust, as in wheels of vehicles, or to prevent contact with water, as in marine work, where the open end of an iron rod, cased with brass, must be covered. In such cases a covered nut is used, of the form shown in Fig. 118. These are generally of cast metal, and are made with a collar to act as a washer. No difficulty need be experienced in designing nuts of this form, they should be made one size larger across the flats than ordinary nuts of the same diameter, owing to their being cast and not wrought, and the depth of the screwed part should equal the bolt diameter. Beyond this room must be left for the end of the bolt to come through for one full thread, then a small clearance, and afterwards the cover, which will vary in thickness from a minimum of about $\frac{1}{8}''$ or $\frac{3}{16}''$. The thickness of the washer may be equal to $\frac{1}{4}d$.

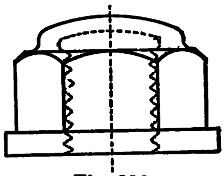


Fig. 118.

EXAMPLES.

SCALE—*Full Size.*

EX. 1.—Draw three views of a hexagonal nut for a $2\frac{1}{4}''$ bolt, not showing the thread.

EX. 2.—Draw three views of a covered hexagonal nut for a $2\frac{1}{2}''$ bolt, one elevation to be in section. Show the end of bolt in position. The thread is to be shown.

EX. 3.—Make drawings of three arrangements for locking a $1\frac{1}{2}''$ bolt and nut. Dimension the parts.

SECTION XVII.

USE AND PROPORTIONS OF BOLTS—STUDS—SCREWS.

Mention has already been made of the difference between a bolt, screw, and stud, and it now remains to point out where each is used, and to generally consider their working conditions and strength.

(35) **Use of Bolts.**—Bolts are always used where possible, on account of the small trouble and time required to fix them.

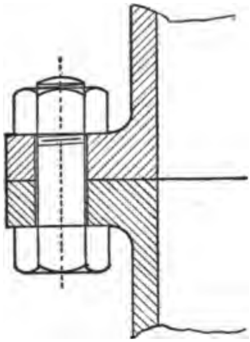


Fig. 119.

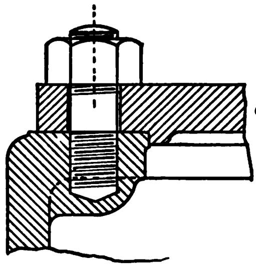


Fig. 120.

Fig. 119 is a good example of the use of a bolt for holding together two flanges, the flange in this case being an *outside* one.

(36) **Use of Studs.**—In Fig. 120 the flange is *inside*, a very common form of construction in engine work, and always necessary for such parts as covers of manholes or handholds. Here it is not possible to use bolts, and *studs* must be employed. This necessitates that the holes in the inside flange shall be screw-tapped, and one end of the studs screwed into the hole up to the plain part. The cover is then dropped over the studs, and the nuts screwed home. The *studs* are, therefore, *fixtures*, and are not removed each time the joint is broken, as in the case of bolts. This is a great advantage, and is often so convenient that studs may be used with outside flanges in order to gain it. It should also be remembered that the use of studs instead of bolts for outside flanges materially reduces the diameter of the

flange, an important consideration where the weight of the parts must be kept down. But many engineers avoid the use of studs wherever possible, owing to their liability to corrode, and to break off across the bottom of the plain part, leaving the screwed end in the hole. This causes much trouble to remove.

It will be seen from Figs. 120 and 121 that an extra piece of metal is cast on the under side of the flange directly beneath the stud, and that the hole for the stud is not drilled right through. This practice is adopted to obtain sufficient length when it is desired to prevent holes being open to the inside of the casting in order to avoid the chance of leaking, as in cylinders, condensers, pump-chambers, &c. Such holes are called "*blind holes*," and are open to objection, owing to the difficulty in tapping them.

(37) **Use of Screw.**—Cases often occur where a pipe or bracket requires fixing to two faces which are, say, at right angles to each other, as in Fig. 121. Bolts cannot be used at

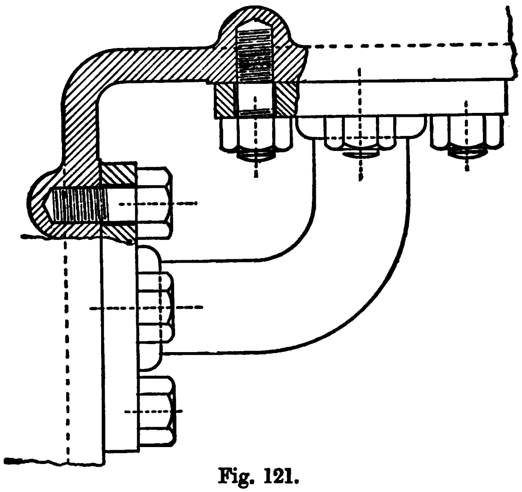


Fig. 121.

either joint, and it will be easily seen that studs could not be fixed in both cases, as it would then be impossible to slip the pipe in position, the studs in one flange preventing the other going on. It is necessary, then, to make one joint with studs (the upper one), and afterwards to use *screws*, as shown, to make the other joint (the lower one). It will be seen that the screw also requires a "tapped" hole in the part to which the pipe is

fixed, and that it is then screwed home until the head bears down tightly upon the flange. Instances may occur where three or more branches require fixing, and where screws must be used for all except one joint.

Studs and screws must never bear against the bottoms of the holes into which they are screwed. If the end of a stud bears against the hole it is not likely to be as tight as if screwed down to the end of the screwed part, while it is evident that a screw must only bear against the underside of its head.

(38) **Clearing and Fitting Holes.**—It is evident that the holes in the flanges of Figs. 119 and 121, and through the cover of Fig. 120, must be slightly larger in diameter, as shown in the drawing, than the bolt, stud, or screw. The holes are then spoken of as "*clearing*" holes, and are generally made $\frac{1}{32}''$ to $\frac{1}{16}''$ larger in diameter than the bolt. Finished bolts may be given a less clearance than "black" bolts. In many cases, as in couplings, connecting-rods, &c., the bolts are made to fit tightly in the holes, and are then distinguished as "*fitting*." It is often necessary to write on the drawing which are "*clearing*" and which "*fitting*" bolts.

Tapped holes, for the use of studs and screws, are generally shown upon a drawing by two circles, the outer one being of diameter equal to the bolt.

The actual fixing of bolts is left to the workman, but it is still necessary that they should be correctly drawn. Hence the student should notice that the length of the plain part of the bolts in Figs. 119, 120, and 121 is in each case less than the thickness of the flange or flanges through which they pass. If this were not so the end of the thread would be reached before the flange, and the joint could not be made tight.

(39) **Proportions of Bolts.**—Bolts of all forms are almost entirely used where they are subjected to purely tensile stress alone, the chief exception being the bolts for shaft couplings, which are subject to shearing. It is, therefore, easy to see in what way their correct proportions may be ascertained. A bolt (as in Fig. 119) might yield by (*a*) pulling apart across the section where the thread ends, (*b*) stripping the thread through the nut by shearing, (*c*) shearing off the bolt head. A stud or screw (Figs. 120, 121) might further yield by stripping the thread from the holes into which it is screwed.

If the student equates the resistance of the material to each of these actions, taking the resistance to stripping as being equal to the resistance to shearing, the nuts and bolt heads will appear to be much thicker than necessary. The actual proportions adopted are the outcome of experiments, during which it was

shown that the threads were injured when the thickness of the nuts were less than seven-tenths of the bolt diameter.

(40) **Strength of Bolts.**—**The strength of a bolt, stud, or screw is taken as being its resistance to breaking across the section at the bottom of the thread.**

Evidently, however, care must be taken to screw a stud or screw sufficiently far into the tapped hole to ensure good holding power. The following rule is generally adopted:—

Studs and screws should be screwed into metal a distance of from $1\frac{1}{4}$ to $1\frac{1}{2}$ times their diameter.

In order that this may be possible, it is often necessary to thicken the part into which the screw or stud screws, by means of "bosses," such as shown in Figs. 120, 121.

It may also be noted that in screwing up a bolt or nut, the friction between the two gives rise to a shearing action on the bolt, due to its tending to turn with the nut.

EXAMPLES.

SCALE—*Full Size.*

EX. 1.—Make a drawing showing one front and two end elevations each of—(*a*) bolt and nut; (*b*) stud and nut; (*c*) screw—of $1\frac{1}{4}''$ diameter, the bolt to be used for two flanges $1\frac{1}{4}''$ thick, and the stud and screw for flanges $1\frac{1}{4}''$ thick (see Figs. 103, 105).

EX. 2.—Draw a front and side elevation and a plan of a $1''$ bolt $3\frac{1}{2}''$ long with nut, when the axis of the bolt is inclined to the ground at an angle of 40°.

EX. 3.—Make drawings to show the different uses of a bolt, stud, and screw, as in Figs. 119, 120, 121, to the following sizes:—

Fig. 119.—Flanges, $\frac{5}{8}''$ thick; other parts, $\frac{5}{16}''$; bolt, $\frac{5}{8}''$ diameter; distance of centre from inside edge of metal, $1\frac{1}{4}''$.

Fig. 120.—Flanges, $\frac{5}{8}''$ thick; stud, $\frac{5}{8}''$ diameter; distance of centre from inside edge of metal, $1\frac{1}{8}''$.

Fig. 121.—Pipe, $2''$ diameter; flanges, $6\frac{1}{4}''$ diameter, $\frac{3}{4}''$ thick; studs and screws, $\frac{3}{4}''$ diameter; centres, $\frac{7}{8}''$ from outside edge of flange; centre of top flange, $4\frac{3}{4}''$ above face of bottom flange; face of top flange, $5\frac{1}{2}''$ from centre of bottom flange.

✓

SECTION XVIII.

FLANGES AND BOLT CONNECTIONS.

(41) Connection by Bolts.—There are a large number of machine parts held together by bolts and nuts, the proportions of which are not altogether determined as questions of strength, but largely depend upon the size of the bolts used.

For instance, suppose a drawing is being made of the connection of two 4″ cast-iron pipes—thickness, $\frac{3}{8}''$; diameter of bolts, $\frac{5}{8}''$; number of bolts and thickness of flanges being known. The draughtsman requires to find the "radius of bolt-circle" and "diameter of flange."

Note.—The former is expressed as a "*radius*," because the workman has to draw the circle upon the flange in marking out the bolts, for which purpose the radius is most convenient; and the latter as a "*diameter*," because that is the size required in turning the patterns of the flange.

The extent of the flange does not affect its strength, and we have only to arrange to get the bolts nicely in.

Let Fig. 119 represent a part section of the flange. There will be a "fillet" where the pipe and flange join of about $\frac{1}{4}''$ radius, and unless this is to be cut away, it is obvious there must be at least $\frac{1}{4}''$ clearance between the pipe and the corner of the nut. Therefore the radius of bolt circle may be :—

$$\text{Radius of pipe inside} + \text{thickness} + \text{clearance} + \frac{\text{nut across corners}}{2}$$

$$\therefore r_b = 2 + \tfrac{3}{8} + \tfrac{1}{4} + \frac{1\frac{5}{16}}{2} = 3\tfrac{5}{16}'' \text{ (to nearest } \tfrac{1}{16}'' \text{ greater).}$$

In the same way the diameter of the flange is arranged to give just enough metal beyond the corners of the nuts to allow a proper finish. This distance varies in ordinary flanges from $\frac{1}{16}''$ to $\frac{1}{4}''$.

The student should notice that if the distance between the bolt holes and the outside of the flange were too small, there would be a tendency for the metal to crack when screwed tightly up. With a proper thickness of flange, however, this is scarcely likely to happen.

Therefore the flange diameter may be :—

$$2\left(\text{radius of bolt circle} + \frac{\text{nut across corners}}{2} + \text{metal beyond nut}\right)$$

$$\therefore d_f = 2\left(3\tfrac{5}{16} + \frac{1\frac{5}{16}}{2} + \tfrac{1}{8}\right) = 8\tfrac{1}{4}'' \text{ (to nearest } \tfrac{1}{8}'' \text{ greater).}$$

(42) Connection by Studs.—We will now take an example of the same kind where a stud is used.

Let Fig. 122 represent a part section of the casting of an ordinary plummer block, P B, the cap C of which is to be held down by $\frac{3}{4}''$ studs. To find the distance between the centres of the studs, and length of cap.

If the holes are drilled too near the edge of the metal, there is a danger of cracking the metal when screwing the stud home, especially if the stud is tight. Therefore the distance (a, Fig. 122) of the stud from the outside edge of the surrounding metal must be sufficient to prevent this.

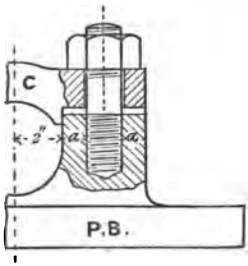


Fig. 122.

Distance of Studs from Edge of Metal.—The rule followed in practice is to allow a distance of not less than half the diameter of the stud. In bearings, cylinder covers, &c., it is generally 0·7 d to d. In the case of tight joints it must be considered together with the question of the width of the joint.

Therefore the distance from centre of bearing to centre of stud may be $2'' + \frac{1}{2} + \frac{3}{8}'' = 2\frac{7}{8}''$, and distance between centres of studs $= 5\frac{3}{4}''$. Allowing the same amount of metal on the other side of the stud, the size across the block will be $2\,(2\frac{7}{8} + \frac{3}{8} + \frac{1}{2}) = 7\frac{1}{2}''$, which is also equal to the length of the cap. Or if we calculate, as in the first example, for the diameter of the flange, we find that a length of $7\frac{1}{2}''$ allows about $\frac{1}{8}''$ beyond the nut, and is, therefore, satisfactory.

Precisely this same method and reasoning applies to nearly all parts connected by bolts and nuts, or studs and screws. Examples will constantly occur in such parts as flanges of pipes and valves, couplings, covers of cylinders and slide chests, connecting-rod ends, caps of bearings, stuffing-boxes, &c. In all these cases the size of the bolt determines the proportions of the parts around the bolt, and the student should now be able to find these proportions in all future work and not require to be told them. Certain other considerations may slightly affect particular cases, but these will be referred to as they occur.

(43) Dimensioning Parts.—In deciding the dimensions of a certain part, some regard must be paid to its general purpose. It may be necessary in intricate machines to work to such small units as hundredths or even thousandths of an inch, but it would be absurd to measure so finely in ordinary work. Few parts of engines or machines require to be measured to less than the

nearest $\frac{1}{16}''$, and in the sizes of flanges, couplings, covers, &c., the nearest $\frac{1}{8}''$ is the smallest unit worked to. The student should now learn to distinguish these points for himself; no good designer, for instance, would think of dimensioning the drawing of an engine or lathe bed as accurately or to as small dimensions as the details of a link motion or slide rest.

EXAMPLES.

Calculate the radius of bolt circle and diameter of flanges and cover in the following examples:—

EX. 1.—Cast-iron pipe $6''$ outside diameter, radius of fillet $\frac{5}{8}''$, diameter of bolts $\frac{3}{4}''$.

EX. 2.—Cover for handhole of $4\frac{1}{2}''$ diameter, diameter of studs $\frac{5}{8}''$.

EX. 3.—Cap for plummer block as in Fig. 122, where the distance marked $2''$ in the figure is $4\frac{3}{4}''$, studs $\frac{7}{8}''$ diameter.

SECTION XIX.

STRENGTH OF BOLTS AND FLANGED JOINTS.

IT has already been stated that the strength of a bolt, stud, or screw is taken to be the tensile strength of the section at the bottom of the thread, and it now remains to consider what shall be the ordinary working load on this section.

Bolts used in flanges or covers require to be screwed up to make the joint tight before the load, due to internal pressure, comes upon them. This screwing up produces a tensile stress in the bolt, which may be sufficient in itself to produce breaking, unless the workman exercises care. For this reason, bolts of less than $\frac{5}{8}''$ or $\frac{3}{4}''$ diameter should not be used for steam-tight joints or for water joints at high pressures. Smaller bolts are often used, but they are always liable to break in screwing up.

It is, therefore, necessary to take a very low value for the stress per square inch on cover bolts due to the internal pressure alone, in order to allow a good margin of safety for screwing up; but apart from this the bolts must be close enough together to prevent the joint leaking between the bolts, and this may require so large a number of bolts as to reduce the load upon them to a very low value indeed.

But this tendency to leak between the bolts may be prevented by increasing the thickness of the flanges or cover, instead of the number of bolts. The student should, therefore, recognise the difficulty of laying down any hard and fast rules, and must

be prepared to consider all the conditions affecting any particular joint. The author has particulars of practical examples of pipe joints for steam pressures of 100 lbs. per square inch, where the stress on the bolts, due to internal pressure, is as low as 1,000 lbs. per square inch, and of cylinder covers for the same pressure, one of which had eight $\frac{3}{4}''$ bolts for a 6″ cylinder, the other having only six bolts of the same size, although the cylinder was $8\frac{1}{2}''$ diameter, the cover of the latter being much thicker than the former, and the greatest stress being 4,000 lbs. per square inch.

The following values will serve as guides for calculation:—

(44) **Stress on bolts due to internal pressure, allowing for stresses due to screwing up.**

Diameter exposed to pressure less than,	6″	10″	15″
Stress on bolts in lbs. per square inch,	2,000	3,000	4,000
Diameter of bolts,	$\frac{5}{8}''$ to $\frac{3}{4}''$	$\frac{7}{8}''$ to 1″	1″ to $1\frac{1}{8}''$

(45) **Pitch of Bolts.—For covers and flanges in steam-engine work, pitch not to exceed five times diameter of bolt.**

For the covers of condensers and air pumps where the pressure is low, or for exhaust steam or water pipes, the pitch may equal six or eight times the bolt diameter.

(46) **Thickness of Covers and Flanges.**—This usually bears some relation to the thickness of the pipe or cylinder barrel, but it is also decided by the size of the bolts used. If the flanges are too thin the metal will crack, due to the stress of screwing up. The following agrees generally with actual practice:—

Thickness of covers or flanges should not be less than the bolt diameter, and is generally equal to it.

(47) **Example.**—The following example will illustrate the method of considering a practical case. It should be noticed we must pay attention to the bolt pitch, as well as to the working load:—

Cylinder 10″ diameter at cover, steam pressure 115 lbs. per square inch. Find size and number of bolts.

Area = 78·5 sq. in. Total pressure = 78·5 × 115 = 9027 lbs.;

∴ total bolt area required at 3,000 lbs. per square inch

$$= \frac{9027}{3000} = 3 \text{ square inches.}$$

$$\text{If } \tfrac{3}{4}'' \text{ bolts, then number} = \frac{3}{0{\cdot}3} = 10$$

$$\text{,, } \tfrac{7}{8}'' \quad \text{,,} \quad \text{,,} \quad = \frac{3}{0{\cdot}42} = 8$$

$$\text{Radius of bolt circle (roughly)} = 5 + \tfrac{3}{4} + \tfrac{3}{8} = 6\tfrac{1}{8}''.$$

∴ circumference = 38·5″, ∴ pitch of $\frac{3}{4}''$ bolts $= \frac{38{\cdot}5}{10} = 3{\cdot}85''$ and $\frac{3}{4}'' \times 5 = 3{\cdot}75''$, so that ten $\frac{3}{4}''$ bolts will satisfy sufficiently the conditions of strength and pitch.

In the same way if eight $\frac{7}{8}''$ bolts are used, the pitch would require to be about 5″, which is greater than five times the diameter. Using nine bolts, or adding to the thickness of cover would meet the case.

EXAMPLES.

Calculate the number and diameter of bolts for connecting the following flanges or covers:—

EX. 1.—Pipes of $8\frac{1}{2}''$ diameter, thickness $\frac{3}{16}''$, steam pressure 100 lbs. per square inch above atmosphere.

EX. 2.—Cover for condenser 9″ diameter of hole, internal pressure 5 lbs. per square inch above atmosphere.

EX. 3.—Cover for slide valve chest, size inside chest $12\frac{1}{4}'' \times 6\frac{1}{4}''$, steam pressure 120 lbs. per square inch above atmosphere.

SECTION XX

PIPES AND PIPE CONNECTIONS.

(48) **Pipes.**—Pipes for the conveyance of steam, gas, or water are made of wrought iron or steel, cast iron, and of copper, the latter being chiefly restricted to marine work.

The thickness of cylindrical pipes to resist bursting may be easily found as follows, when the maximum pressure, diameter, and resistance of the material is known:—

Let d_p = internal diameter in inches (when the thickness is great, d_p should equal the mean diameter).
l = length of pipe or cylinder, neglecting flanges, in inches.
t = thickness of pipe or cylinder in inches.
P = maximum internal pressure in lbs. per sq. inch.
f_t = safe stress in the material in lbs. per sq. inch.

Then the bursting effort is equal to $P \times d \times l$, and this is resisted by the tensile strength of the two sections of the shell, each being of length l and thickness t, the total resistance being $2\,(l \times t \times f_t)$.

$$\therefore 2\,(l\,.\,t\,.\,f_t) = P\,.\,d_p\,.\,l.$$

$$\therefore t = \frac{P\,.\,d_p}{2\,f_t}.$$

In actual engineering practice the thickness of pipes, especi-

ally when made of cast iron, is always in excess of the thickness, calculated in this way in order to meet the requirements of manufacture.

(49) **Iron and Steel Pipes.**—Wrought-iron or steel pipes or tubes are generally manufactured as a speciality, and are obtained by the users from the makers who guarantee them of given strength. They are usually made in lengths of from 12 to 16 feet.

Tubes of 12″ diameter may be used with a working pressure of 500 lbs. per square inch, the pressure increasing as the diameter decreases. It is, therefore, sufficient for the purposes of this work to notice only the following proportions :—

Diameter in inches,	1 to $2\frac{3}{4}$	3 to $\frac{1}{4}$	$4\frac{1}{4}$ to 6	7
Thickness in inches,	$\frac{5}{32}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{1}{4}$

(50) **Cast-Iron Pipes.**—The thickness of cast-iron pipes is a question of foundry possibilities rather than one of theoretical strength. It is impossible to cast thin pipes with any degree of uniformity, the result being that cast-iron pipes and cylinders are invariably much thicker than theoretically necessary. A workable thickness is given by taking the greatest resistance of the material as equal to 1,800 lbs. per square inch, taking care that the minimum thickness is not less than $\frac{5}{16}$ for short lengths, and $\frac{7}{16}$ for long lengths. The following are practical proportions for pressure up to 250 lbs. per square inch :—

Diameter in inches,	2 to 6	7	8	9	10
Thickness in inches,	$\frac{3}{8}$ to $\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$

(51) **Copper Pipes.**—Copper pipes are made from malleable sheets and may be as thin as $\frac{1}{16}$″. The thickness for any diameter and working pressure may be found by taking the greatest resistance of the material as equal to 3,000 lbs. per square inch. The following are practical sizes for pressures up to 100 lbs. per square inch :—

Diameter in inches,	1 to 6	8 to 12	12 to 18
Thickness in inches,	$\frac{1}{16}$ to $\frac{3}{16}$	$\frac{1}{8}$ to $\frac{1}{4}$	$\frac{1}{4}$ to $\frac{3}{16}$

The thickness of copper sheets is usually measured with the Birmingham Wire Gauge.

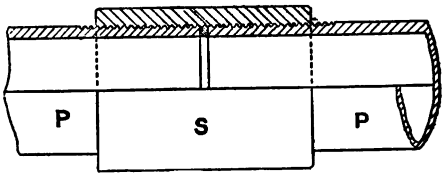


Fig. 123.

(52) **Pipe Connections.**—Wrought-iron or steel pipes are often connected by a simple screwed socket, as shown in Fig. 123. The pipe ends P P and the socket S are screwed with right- and left-handed

threads, so that the two pipes are drawn closely together until the ends meet, a ring of packing being often inserted between them. The socket should embrace a length of each pipe equal to from $1\frac{1}{4}\ d_p$ to $1\frac{1}{2}\ d_p$, and in thickness is equal to one and a-half times the pipe thickness.

Figs. 124*a, b, c, d*, show standard forms of pipe connections.

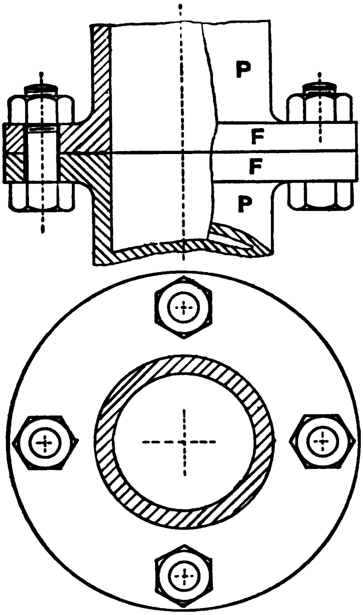


Fig. 124*a*.

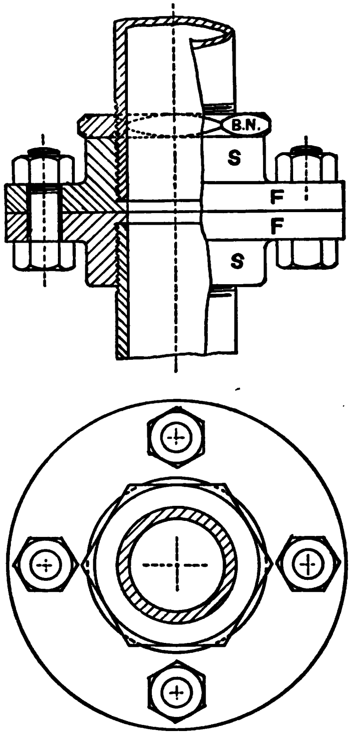


Fig. 124*b*.

(53) Cast-Iron Pipes (*a*).—Here the flanges F F are cast with the pipes P P, a "fillet" of radius equal to the pipe thickness up to a maximum of $\frac{1}{2}''$ being cast at the junction of the flange and pipe. The flange thickness equals from 1·3 to 1·5 times the pipe thickness. As this is in excess of the thickness required from considerations of bolt diameter, the bolts used for cast-iron pipe flanges are generally smaller in diameter than the thickness of the flange.

(54) **Wrought-Iron Pipes** (*b*).—Flanges, F F, of cast iron made thicker next the pipe by a spigot, S, in order to give a greater length of thread. Some makers are satisfied with simply screwing the flange tightly on, as in the lower pipe, others use "back nuts," B N, as shown on the upper pipe, which jamb against the back of the spigot, and make a more rigid joint. These nuts are better octagonal when of large size.

The total depth of the flange and spigot is made about one and a-half to one and three-quarter times the flange thickness, when as on the lower pipe. In the upper pipe the depth should not be less than two and one-quarter times the flange thickness, in order to keep the back nuts clear of the flange-bolts and nuts. The back nuts do not follow the sizes of ordinary nuts. They need not exceed a thickness of $\frac{3}{8}''$ to $\frac{5}{8}''$, and a size across the flats arranged to give a minimum width of metal of $\frac{3}{8}''$ to $\frac{9}{16}''$. The spigot diameter depends upon the nut, and should not be less than the size across the flats of the nut, due regard being paid to sufficient thickness of metal. Thickness of flanges need not be less than bolt diameter.

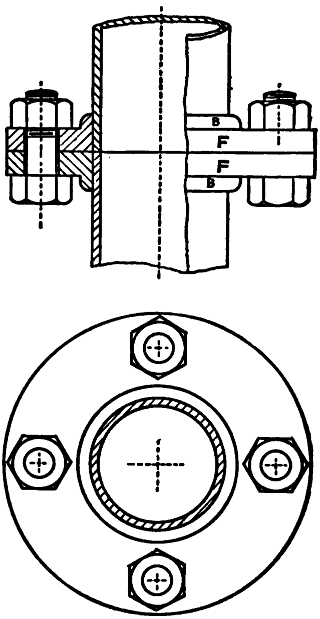


Fig. 124c.

(55) **Copper Pipes** (*c*).—Flanges, F F, of gunmetal, made thicker by a beading, B, next the pipe, the flange and pipe being brazed together. Beading for pipes up to 8″ diameter, from $\frac{1}{4}''$ thick × $\frac{5}{16}''$ deep to $\frac{5}{16}''$ thick × $\frac{3}{8}''$ deep. The flange thickness may be from three to four times the pipe thickness, and should not be greater than the bolt diameter.

It is important to note that, although the general aim in designing flanges of pipes and valves is to keep the diameter as small as possible, yet in cases where there is a long range of horizontal piping, the flanges serve materially to stiffen the pipes and to prevent buckling. In such a case the diameter is often made larger than usual, and similarly with a valve having one flange bolted

to a cylinder or condenser and the other flange supporting a line of piping where a rigid joint is very necessary. Also, the question of how near the bolts may be placed to the pipe is modified by whether the pipe is easily getatable, or placed where it is difficult to use a spanner. These points should always be looked to.

(56) **Union Joint** (*d*).—This joint is used for small brass and copper pipes, where it may be necessary to frequently break and make the joint, as in connections to pressure gauges, portable pumps, &c. Each pipe is screwed fast into a socket, A and B, of which A is made with a screwed end and a nut part, N, for holding with a spanner, and B with a small collar or flange. A

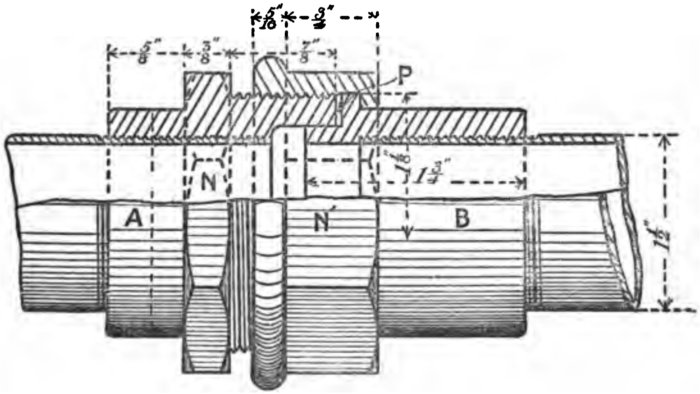


Fig. 124*d*.

loose, partly covered nut, N′, forms part of the connection B, which, screwing on to the screwed end of A, draws the two pipes tightly together against a ring of packing placed between the ends of A and B in the space marked *p*. The connections are of cast brass, and the nuts are made octagonal for compactness, and of a size across the flats equal to some size of Whitworth standard nut.

(57) **Elbows and Tee Pieces.**—Lengths of piping are connected together at different angles by means of short bent lengths called "elbows," "bends," or "tee pieces." An elbow or bend has usually two flanges at an angle to each other, and when fixed in a line of piping alters the direction of the line according to the angle. A tee piece has three flanges, and really consists of a short length of pipe connected at right angles to another length, and, therefore, resembling the letter **T**. It is

used to lead off a line of piping at right angles to and from any part of another line.

(58) **Order of Drawing.**—In drawing pipe connections it is sufficient to show two views only, as in Fig. 124, and it is also convenient to section one-half of the elevation. When pressed for room, it may suffice to show only one-half of the plan. The order of drawing should be as follows:—

(1) Centre lines of both views.

(2) Lines showing inside and outside diameter.

(3) Lines showing thickness of flanges and spigot in elevation.

(4) Draw-in "fillet" of flange, and spigot (decide size of nut across flats).

(5) Decide position of bolts, and show by centre lines in both views.

(6) Bolts and nuts in both views.

(7) Decide diameter of flange, and show in both views.

N.B.—Arrange the position of the bolts, so that one is seen fully in the sectional elevation as in the figures.

EXAMPLES.

Make working dimensioned drawings to a scale of 6″ = 1′ of the following pipe connections showing part of the elevation in section. Steam pressure 100 lbs. per sq. in. above atmosphere (see Fig. 124):—

EX. 1.—Cast-iron pipes 5″ diameter, thickness $\frac{7}{16}$″.

EX. 2.—Wrought-iron pipes 4″ diameter, thickness $\frac{3}{16}$″ with spigots and back nuts.

EX. 3.—Copper pipe 6″ diameter, thickness $\frac{3}{16}$″.

EX. 4.—Copper pipe $1\frac{5}{8}$″ diameter, thickness $\frac{3}{32}$″ full, connected with union joint, as shown in Fig. 124 *d*.

SECTION XXI.

CONNECTION OF RODS AND SHAFTS—PIN JOINTS—COTTERED JOINTS—KEYS—COUPLINGS.

Rods and shafts may be connected together with either working or fixed joints, the former allowing independent movement of the rods, and generally effected by some form of pin joint, the latter rigidly connecting the rods as though of one piece, and effected by a "cottered joint" or "coupling."

(59) **Knuckle or Forked Joint.**—This is a common type of pin joint, and in the form shown in Fig. 125 is used in both machines and structures. It consists of two parts, a "forked or double eye" shown at A, and a "single eye" shown at B which fits between the fork of the double eye, the two parts being connected by a turned pin passing through both, and shown in position in A.

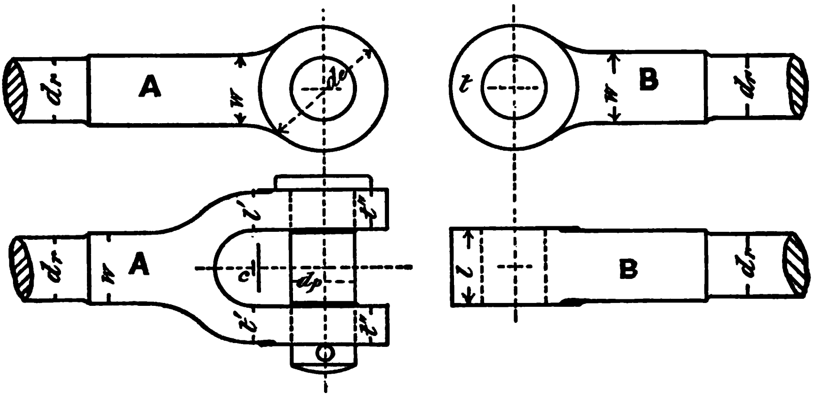


Fig. 125.

In designing such a joint the proportions of the pin and eyes are made to be of equal strength with the rod. The joint is used for both tensile and compressive forces, but as it is only made in wrought iron or steel it may be regarded as subject to tension only. Then by taking the lettering of the figure, the joint may fail as follows:—

I. By breaking through the section of the rod d_r.

$$\text{Resistance to tension} = d_r^2 \times \frac{\pi}{4} \times f_t.$$

II. By shearing the pin across the two sections marked by dark lines.

$$\text{Resistance to shearing stress} = 2\left(d_p^2 \times \frac{\pi}{4} \times f_s\right).$$

III. By the pin shearing through the end of the single eye; thickness t, and length l.

$$\text{Resistance to shearing stress} = 2\,(l \times t \times f_s) \text{ (approximately)}.$$

IV. By breaking through the two sections of the fork of the double eye; thickness t', and width w. There is also a bending action tending to open the fork when the joint is subject to compressive stresses.

$$\text{Resistance to tension} = 2\,(w \times t' \times f_t).$$

If I. and II. are equated, taking $f_s = \frac{4}{5} f_t$, then the diameter of the pin $d_p = 0{\cdot}79\, d_r$.

In the case of III. either l or t must be assumed in terms of d_r. It will be pointed out later that l must slightly exceed d_r, but taking them as equal then $t = 0{\cdot}5\ d_r$.

Evidently then the thickness t'' of each eye of the fork, and, therefore, the thickness t' of the fork must not be less than $\frac{1}{4}\ d_r$, or the pin would shear through the double eye; also for practical reasons w is not less than d_r, and hence the resistance to tension of IV. is $2\ (d_r \times \frac{1}{2}\ d_r \times f_t) = d_r^2 f_t$, which is greater than the resistance of the rod. It is, therefore, unnecessary to consider the proportions of the fork, as they naturally follow with ample strength from the other proportions.

The usual proportions adopted in practice exceed these, in order to allow for any bending stresses produced when the pin wears and no longer fits tightly in the holes. These proportions are as follows:—

Diameter of pin d_p $= $ diameter of rod d_r.
Diameter of eye d_e $= 2\ d_r$.
Thickness of eye t'' $= \frac{3}{8}\ d_r$.
Length of single eye $l = 1\frac{1}{4}\ d_r$.

All other proportions can be obtained from these by simply allowing for the parts to be machined, and taking care that the machining of one part does not interfere with the machining of any other part. It is because of this that the size w exceeds the diameter of the rod, and the thickness of the fork in the double eye t' is less than the thickness of the eye t''. The thickness of the single eye l also exceeds the diameter of the rod d_r for the same reason.

The centre c for the curve of the fork should be taken as shown, just beyond the single eye, to ensure flat surfaces between the eye and the fork. The centre for the outer arcs are often taken still further to the left of c, in order to make the fork thicker where it joins the rod to better resist bending.

The parts of the joint marked w, and shown of square section, may be finished octagonal at the ends near the rods, and in length may equal $2\ d_r$.

(60) **Pin for Knuckle Joint.**—There is no stress on the pin-head, which may have a diameter of $1\frac{1}{2}\ d_p$ and a thickness of $\frac{1}{4}\ d_p$, to a minimum of $\frac{1}{4}''$. A taper keep pin should be fitted as shown.

N.B.—Attention may be paid here to the best radius for the curved part of the double eye where the rod and fork join. Such a construction is exceedingly common, and the radii adopted are arranged to give a good sense of proportion and finish. As a general rule, it is convenient to draw both curves with the same radius; this, of course, being modified if the part then looks heavy or light.

A joint identical in construction to the double eye of the knuckle joint is extensively used for the crosshead end of con-

necting-rods, and for the end of eccentric-rods where they are attached to the valve-link motion, and will be referred in the sections on engine design. A rough forged joint of the same form is also commonly applied in boiler, roof, and bridge work.

(61) **Common Pin Joint.**—A simpler form of the knuckle joint is used in small engines for connecting the eccentric- and slide valve-rods, and is shown in Fig. 126, E R being the eccentric-rod, and V R the valve-rod. As such a joint is alternately subjected to tensile and compressive stresses, it is considered advisable to fit a small wedge or brass packing piece between the single eye and the back of the fork, as shown in the figure, as this prevents knocking when the pin wears loose. The pin is frequently made without a head, and held in position by a keep pin, as shown.

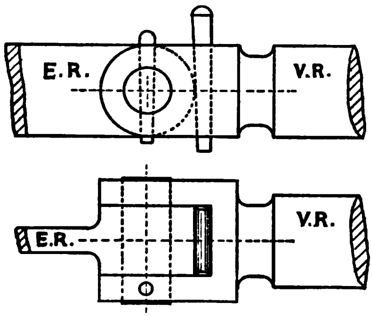


Fig. 126.

As the section of the rods to be connected by the pin joint is determined by other considerations than as a case of mere tension, the proportions of the joint must not be obtained from the rod, but from the actual tensile and compressive force acting upon the rod. For example, suppose the greatest tensile stress on such a joint as in Fig. 125 is known to be 3,900 lbs. Then considering tension only, the rod diameter d_r for a working stress of 5,000 lbs. per sq. inch, would require to be 1″, which may, therefore, be taken as the standard for the other proportions. The increased length of the single eye in Fig. 126 is to give sufficient wearing surface on the pin, and will be referred to again later.

(62) **Drawing Pin Joints.**—It is enough to show two views, a plan and elevation. The joint should be drawn together, the pin being shown by dotted lines in order to avoid a section. The drawing should proceed in the order in which the different proportions have been shown to follow from the diameter of the rod.

EXAMPLES.

Make full-size dimensioned drawings of the following pin joints:—

EX. 1.—Knuckle joint for rods of $1\frac{1}{4}''$ diameter (Fig. 125).

EX. 2.—Joint for eccentric and valve-rod—pin, $\frac{3}{4}''$ diameter; eccentric-rod, $1\frac{1}{2}'' \times \frac{1}{2}''$; valve-rod, $1\frac{3}{8}''$ diameter; distance between jaws of double eye, $1\frac{1}{4}''$ (Fig. 126). Total tensile stress on rod = 3,000 lbs.

(63) **Cottered Joints.**—Cottered joints are used to rigidly connect rods which transmit motion and do work in the direction of their length, without rotation, as in the pump-rods of mine shafts or wells, or as commonly seen in steam engines, to connect the piston-rod and crosshead, or different lengths of the slide valve-rods.

When used to connect two rods, one of the rods is made with a socket end, B, into which the other end, A, fits (Fig. 127), or a separate sleeve fits over the end of each rod. The cotter C is a thin taper bar driven tightly through the box and rod, as shown.

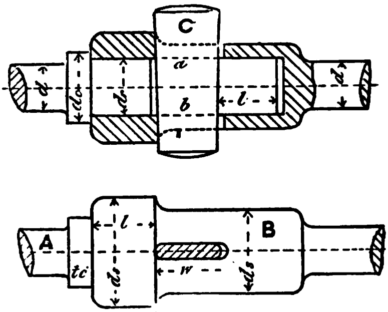


Fig. 127.

Cotters taper in width, not in thickness, the usual total taper being 1 in 24, or $\frac{1}{2}''$ per foot. They thus act as wedges, and draw the parts closely together until the collar on rod A jambs tightly against the socket B. The bearing edges should be made semicircular, as seen in the plan, as this gives a better surface than square edges, and allows the cotter holes to be drilled.

(64) **Clearance of Cotters.**—Notice that in order to draw the two rods A and B together, the cotter must only bear against the rod A on the right-hand edge, and the rod B on the left-hand edge; hence there must be clearance between the left hand of rod A and the cotter, and also between the right hand of rod B and the cotter. In this form of joint, where the cotter is driven right home at the time of fitting, this clearance need not exceed $\frac{1}{8}''$.

(65) **Strength and Proportions of Cottered Joint.**—It is evident that when the rods are subject to tensile or compressive stresses, there is a shearing action on the two sections of the cotter at *a* and *b*. The cotter area which resists this at each section is $w\ t$ sq. ins., where w = width of cotter and t = thickness. One of these two sizes must be agreed upon to commence with, so that we have the general rule.

Thickness of cotter = $\frac{1}{4}$ d_r where d_r is the diameter of the rod through which the cotter passes.

Cotters are best made of mild steel, and the following calculations apply to the usual combination of wrought-iron rods and a steel cotter. For other metals it is only necessary to change the value of the shearing strength of the cotter and the tensile strength of the rods, according to the metals employed.

Let d = diameter of rods.
d_r = diameter of rod through which cotter passes.
d_c = diameter of collar on rod A.
d_s = outside diameter of socket B, into which d_r fits.
t_c = thickness of collar on rod A.
t = thickness of cotter.
w = width of cotter.
l = length of rod or socket beyond the cotter.
f_s for steel cotter $= f_t$ for wrought-iron rods.

Assuming the joint to be in tension, it may yield in one of the following ways :—

I. By pulling apart the rods A or B.

$$\text{Resistance to tension} = d^2 \times \frac{\pi}{4} \times f_t.$$

II. By pulling apart the rod d_r through the cotter way.

$$\text{Resistance to tension} = \left\{ \left(d_r^2 \times \frac{\pi}{4} \right) - d_r\, t \right\} f_t$$

III. By shearing through the cotter at the sections a and b.

$$\text{Resistance to shearing stress} = 2\,(w \times t) f_s$$

IV. By pulling apart the socket through the cotter holes.

$$\text{Resistance to tension} = \left\{ \frac{\pi}{4} \left(d_s^2 - d_r^2 \right) - (d_s - d_r)\, t \right\} f_t.$$

V. By compressing the rod d_r or the socket where the cotter bears.

$$\text{Resistance to compression of rod} = (d_r \times t) f_c$$
$$\text{Resistance to compression of socket} = t\,(d_s - d_r) f_c$$

VI. By compressing the collar d_c where it bears against the socket.

$$\text{Resistance to compression} = \frac{\pi}{4} (d_c^2 - d_r^2) f_c$$

VII. By shearing the collar d_c off the rod A.

$$\text{Resistance to shearing stress} = (d_r \times \pi \times t_c) f_s$$

VIII. By shearing through the rod or socket beyond the cotter.

$$\text{Resistance to shearing of rod} = 2\,(l \times d_r) f_s$$

$$\text{Resistance to shearing of socket} = 2 \left\{ l\,(d_s - d_s) f_s \right\}$$

In VII. and VIII. the value of f_s should be taken as equal to $\frac{2}{3} f_t$, because the shearing is parallel with the fibres, and offers less resistance than to transverse shear.

By equating each of these resistances, II. to VIII., to the tensile resistance of the rod, as in I., and substituting the equivalent value of f_s in terms of f_t, and, also, where necessary for t in terms of d or d_r, and for d in terms of d_r, the following proportions will result:—

From II. $d_r = 1{\cdot}2\,d$.
,, III. $w = 1{\cdot}3\,d$.
,, IV. $d_s = 1{\cdot}7\,d$.
,, V. $d_s = 2{\cdot}4\,d$ to give equal bearing pressure $f_c = 2f_t$ on both rod and sleeve.
,, VI. $d_c = 1{\cdot}4\,d$.
,, VII. $t_c = 0{\cdot}5\,d$.
,, VIII. $l = 0{\cdot}8\,d$.

In practice the following proportions are generally adopted for steel cotter and wrought-iron rods, where d = diameter of rod, the sizes being dimensioned to nearest $\frac{1}{16}''$ greater.

Cotter—width $= 1{\cdot}3\,d$; thickness $= 0{\cdot}3\,d$.
Collar on rod—diameter $= 1{\cdot}5\,d$ thickness $\doteq \frac{1}{2}\,d$.
Diameter of rod through which cotter passes $= 1{\cdot}2\,d$.
Outside diameter of socket $= 2{\cdot}4\,d$ in front of cotter.
Cotter not to be nearer end of rod or socket than $1\frac{1}{4}\,d$.

In order to avoid a socket thicker than necessary for its resistance to tension, and yet thick enough at the cotter hole to stand the bearing stress, the socket may be made of a diameter equal to $1{\cdot}7\,d$, and have a top and bottom lug, as in Fig. 128, to

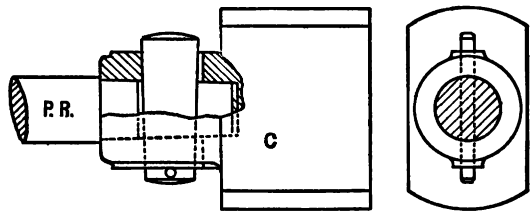


Fig. 128.

make up the necessary thickness. This is a very good arrangement, but is expensive to adopt except for cast-iron sockets. An equally good method is to reduce the socket diameter to $1{\cdot}7\,d$ behind the bearing edge of the cotter, as shown in Fig. 128.

(66) **Forms of Cottered Joints.**—A common application of cottered joints in steam engines is to rigidly connect the piston-rod and crosshead. Such a joint is shown in Fig. 128. The end of the piston-rod P R is tapered, thus dispensing with a collar. Total taper of rod, 1 in 12 to 1 in 20. When the crosshead C is of cast iron, as is frequently the case, the proportions of the socket part through which the rod passes must be arranged

accordingly, and it is better to take a definite value for the total tensile stress on the rod, and for f_t, f_c, and f_s, because a piston-rod is abnormally strong when calculated only for the tensile and compressive stresses upon it, in consequence of the rod having to resist buckling. It is because of this that the rod may be decreased in diameter at the tapered part, and it will be in accordance with practice to take the tensile strength of the tapered part through the cotter way as the resistance of the joint and make all other resistances equal to it.

Another form of this joint is shown in Fig. 129, where the rod

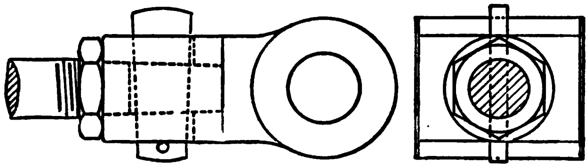


Fig. 129.

is screwed and a nut (thickness = $\frac{1}{2}$ d) is used which jambs up against the socket after the cotter is driven home, thus taking up any possible slack and helping to release the rod and socket when disconnecting.

Split-pins should be put through the bottom end of the cotters as shown in the figure to prevent them working loose.

The length of the cotter need not exceed the outside diameter of the socket by more than about 1″.

(67) **Gib and Cotter.**—When a cotter is used for the purpose of connecting a thin strap and a comparatively thicker rod, as in the case of some engine connecting-rods (see Fig. 191), the cotter is made about half the usual width and is used with a second hooked cotter called a "gib," the top and bottom projections of which hook over the strap and thus prevent it opening. This arrangement is shown in Fig. 130, where R is the rod, S the strap, C the cotter, and G the gib.

The gib and cotter are parallel along their outside edges and taper on their inside edges. In such cases a small set screw S′ ($\frac{3}{8}$″ to $\frac{5}{8}$″), is fitted, screwing through the rod and jamming against the cotter, and prevents it loosening, or if this is not possible the gib is extended by a screwed rod fastening the key with top and bottom nuts; diameter of screw equal width of key, length of screw to allow of top nut being screwed fully on when key is just home. Both these are shown in the figure although only one method is required.

Width of gib and cotter = 1·3 d as before, each being half this width at the centre line.

Thickness of Cotter = $\frac{1}{4}$ d_r, where d_r equals width of strap.

Taper may equal 1 in 12 when screw fastening is used.

Height of head and gib and overlap = $1\frac{1}{8}$ t.

Clearance not less than difference in width of cotter at widest part, and at part where it enters the strap.

Cotter holes in strap not nearer to edge than 0·7 w, where w = width of gib and cotter.

The taper of rods and cotters are usually dimensioned to the nearest $\frac{1}{32}$″.

Cotters are often used as fastenings in such cases as foundation and holding-down bolts for machines and engines, where it is not possible or convenient to use an ordinary bolt or stud. An example is shown in Fig. 131, where the bolt is dropped down

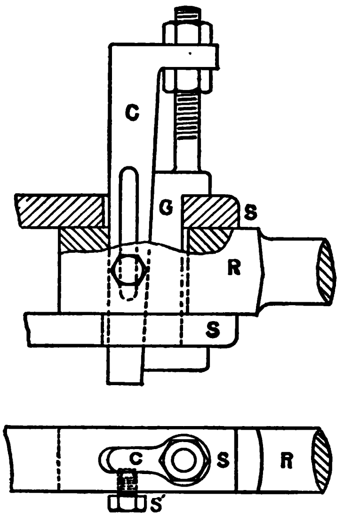


Fig. 130.

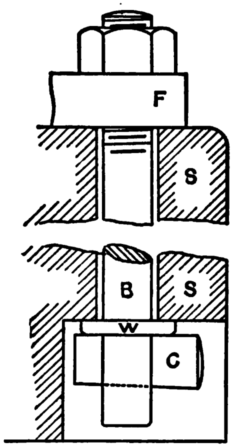


Fig. 131.

from above the cotter driven in from the side, and the whole tightened by screwing down the nut. F is the flange to be bolted down, S the stone bed, B the bolt, C the cotter, and W a washer.

(68) **Drawing Cottered Joints.**—In drawing a cottered joint it is best to show an elevation and plan, the latter being partly in section or clearly shown by dotted lines. It is convenient to draw the cotter by showing the mean width on the centre line, and then tapering it above and below.

EXAMPLES.

EX. 3.—Draw a cottered joint full size and showing two views, as in Fig. 127, for connecting-rods of 2″ diameter.

EX. 4.—Draw a cottered joint full size, and showing two views for a holding-down bolt for foundations, as in Fig. 131. Diameter of bolt $1\frac{1}{2}''$, end of bolt for cotter enlarged and made square, so that effective area is not diminished by cotter way.

KEYS AND COUPLINGS.

Keys are parallel or wedge-shaped pieces used to fix wheels and pulleys to shafting, or to fix couplings on shafts when two or more require to be connected together.

The parts connected by keys invariably transmit motion by rotation, and keys are, therefore, subject to shearing and crushing stresses.

(69) **Feather Key.**—When the wheel or pulley requires to turn with the shaft, and also to move along the shaft, the key is simply a parallel strip called a "feather," sunk for half its depth in the shaft, the other half fitting in a slot or key way cut in the wheel boss, see Fig. 132, where S is shaft, B boss of wheel, and F the feather.

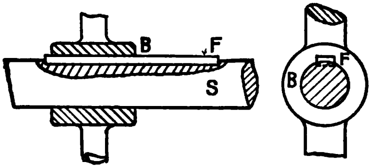


Fig. 132.

(70) **Ordinary Keys.**—When the wheel or coupling requires to be rigidly fixed to the shaft, as is generally the case, the key is usually slightly tapered in the direction of its thickness only, and is driven tightly into place, half being sunk in the shaft and half in the wheel boss or coupling (Fig. 133, *a*). The thicker end of the key terminates in a head, which enables the key to be drawn out of position for disconnecting (Fig. 133, *b*).

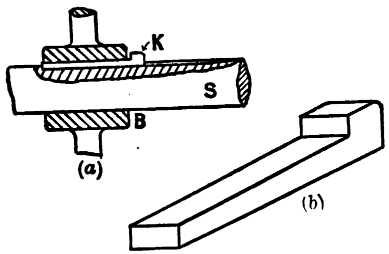


Fig. 133.

(71) **Strength and Proportion of Keys.**—When length of key $= l$, width $= w$, thickness $= t$; then the resistance of the key to shearing $= l \times w \times f_s$ and the resistance to compression $= l \times \frac{1}{2}t \times f_c$ approximately. Evidently, then, the strength

of a key varies directly with its length, and ought to be proportioned according to the work which the wheel or coupling in which they are fitted is called upon to do.

This result appears to be overlooked in practice, except indirectly in the fact that keys are made equal in length to the wheel boss, or coupling into which they fit, and that the length of the boss or coupling bears some relation to the work done, and, therefore, to the shaft diameter.

The following proportions are adopted in practice where d_s = diameter of shaft:—

Taper of keys = 1 in 64 to 1 in 100, or $\frac{1}{8}''$ per foot.
Width of key $w = \frac{1}{4}\,d_s$ to $\frac{1}{4}\,d_s + \frac{1}{16}$ (minimum of $\frac{1}{4}''$).
Thickness of key when sunk $t = \frac{1}{2}\,w$ (minimum of $\frac{3}{16}''$).
Length of head $= \frac{3}{4}\,w$; height above key $= \frac{3}{4}\,t$.

Practical Rule.—The length of wheel bosses, or couplings, keyed on shafts should not be less than $1\frac{1}{4}\,d_s$ for steel keys and $1\frac{3}{4}\,d_s$ for iron keys, with an ordinary single key (two or more keys may be fitted when sufficient strength cannot be obtained with one).

That such a length of key gives ample strength is seen from the following calculation, which makes the resistance of the key equal to the resistance of the solid shaft, and, therefore, supposes the whole of the forces in the shaft to be transmitted by the key. This is only the case in shaft couplings and main driving wheels, as in ordinary workshop shafting only a small part of the work done by the shaft is taken off at each pulley:—

Let l = length of key, $w = \frac{1}{4}\,d_s$, $t = \frac{1}{8}\,d_s$.

Then (I) Resistance of key to shearing = Resistance of shaft to torsion.

$$(l \times w \times f_s)\,\frac{d_s}{2} = 0\,196\,d_s^3 f^*$$

$\left(\frac{d_s}{2}\right.$ is the radius of the shaft, and is, therefore, the distance from the shaft centre at which the key offers a resistance)

expressing w in terms of d_s, $\therefore\ l = 1{\cdot}57\,d_s \times \frac{f}{f_s}$.

If $f = 9{,}000$ lbs. and $f_s = 8{,}000$ lbs. for wrought iron, and 10,600 lbs. for steel, then

$l = 1{\cdot}76\,d_s$ for wrought-iron keys.
$l = 1{\cdot}33\,d_s$ for steel keys.

---

* The resistance of a shaft to torsion is given by $\frac{\pi}{16}\,d_s^3\,f$, or $0{\cdot}196\,d_s^3 f$, where d_s = diameter, and f = stress per square inch in outside fibres. The expression $\frac{\pi}{16}\,d^3$ is known as the modulus of the section for torsion. The value of f is usually taken at 5,000 lbs. for wrought iron and 6,000 lbs. for steel, in order that the twist of the shaft may not be excessive.

(II) Resistance of key to compression = Resistance of shaft to torsion.

$$l \times \frac{t}{2} \times \left(\frac{d_s}{2} + \frac{t}{4}\right) f_c = 0{\cdot}196\, d_s^3 f$$

expressing l and t in terms of d_s $\left(\frac{d_s}{2} \times \frac{t}{4}\right.$ = distance of centre of key from centre of shaft).

$$f_c = 1{\cdot}68 f.$$

When $f = 9{,}000$ lbs., then $f_c = 15{,}120$ lbs. square inch, which is well within the compressive strength.

(72) **Box Couplings.**—The most common method of connecting lengths of shafting for general driving purposes is by means of "box or muff couplings," shown in Fig. 134.

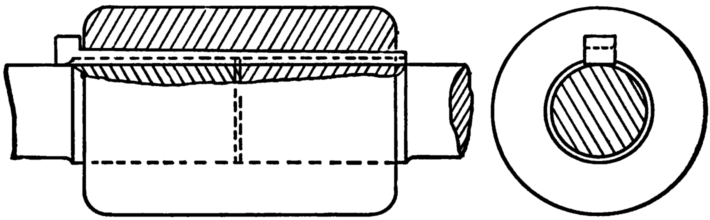


Fig. 134.

The coupling is a simple sleeve, generally of cast iron, fitting for half its length over each of the shafts, the ends of which should not butt closely against one another in order to allow for expansion. Either a key is driven in for the whole length, or one equal to half the length of the coupling may be driven in from each end.

Unless the shaft possesses a good margin of strength it is not good practice to sink a key into the shaft without first increasing its diameter. When this is done as in Fig. 134 it is also much easier to fit the key, for it will be seen that without it it is necessary to cut a long key way (see Fig. 133, *a*) in order to drive the key home. The diameter need only be increased to allow for half the thickness of the key—that is, when d_s = diameter, and $\frac{d_s}{8}$ = thickness, then $d = 1\frac{1}{8}\, d_s$. The disadvantage of enlarging the shaft ends is that ordinary pulleys cannot be slipped over the ends.

The usual proportions adopted in practice are as follows :—

Length of coupling = 3 to $3\frac{1}{2}$ times diameter of shaft.
Outside diameter of coupling = $1\frac{3}{4}$ to 2 times diameter of shaft.

The length is arranged to give sufficient length of key, and the diameter to ensure that the box will not split for its whole length through the key way due to the tensile stress produced by driving home the key and by the work done. The proportions give ample strength.

(73) **Friction Coupling.**—Fig. 135 shows the arrangement of Butler's patent coupling, which dispenses with keys, and depends for its holding power upon the frictional grip of the cone bushes C C. These bushes are split along one side to leave a space, marked S, when driven home, and have a usual total taper of 1

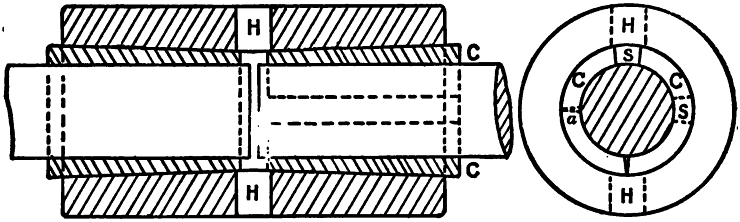


Fig. 135.

in 17. The coupling is rapidly connected by hammering home the bushes, and as quickly disconnected by simply passing a key-drift through the space, marked S, left by one sleeve, and driving against the end of the other sleeve; H H are holes through the coupling. When the bushes are properly driven into place, the holding power of the coupling is very great.

(74) **Flange Couplings.**—Flanged couplings are either made of cast iron and keyed to the shaft, or forged with the shaft as one piece. The former is most common for ordinary driving purposes, and the latter for large size crank and propeller shafts of marine engines.

The connection of cast-iron flanged couplings is shown in Fig. 136. The bolt heads and nuts are recessed into the flanges to avoid projections which would be likely to catch the clothes of the workmen when oiling or putting on belts, and the keys are driven in from the shaft end, and finished off flush, without heads. When the coupling is fixed, quite out of ordinary reach, the bolt heads and nuts need not be recessed, but may stand out beyond the flanges, as in Fig. 137. The faces of the flanges are frequently partly recessed in casting from $\frac{3}{8}''$ to $\frac{1}{2}''$, as shown, to lessen the amount of turning required. The collars or bosses on the flanges are necessary to give sufficient length of key. Bolts to be "fitting."

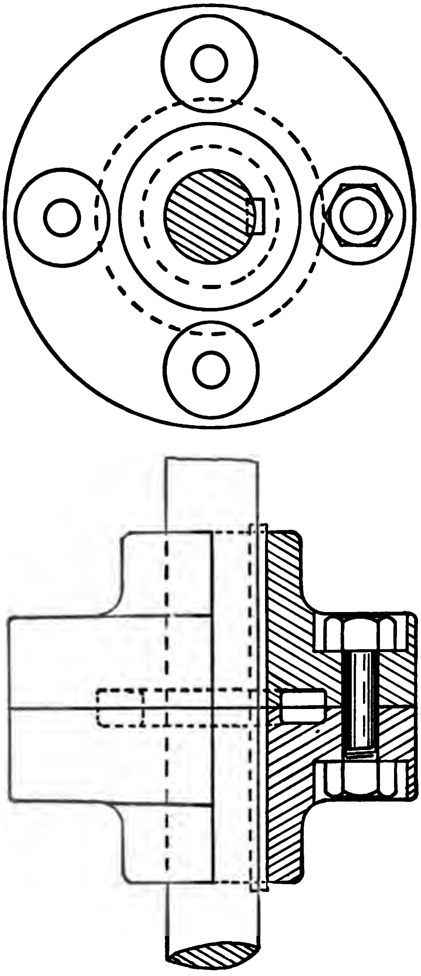


Fig. 136.

The following are proportions adopted in practice :—

Length of flange and boss = $1\frac{3}{4}\ d_s$ (this is to give sufficient length for the keys).

Diameter of boss = $2\ d_s$.

Total thickness of flange when recessed = $0{\cdot}4\ d_s + \frac{1}{16}''$ thickness of nut.

Thickness of flanges when not recessed = 0·4 to 0·5 d_s.
Diameter and number of bolts (see below).
Diameter of recess for bolt heads and nuts must be enough to allow of a box spanner being inserted over the nuts, and may be made = 2 d_s + $\frac{1}{8}''$ to 2 d_s + $\frac{5}{8}''$.
These couplings are often fitted with the end of one shaft projecting into the coupling of the other shaft, a distance of $\frac{1}{8}''$ to $\frac{1}{4}''$, to assist in setting the shafts in line.

The student should have no difficulty in deciding the radius of bolt circle and diameter of flange after what has been said in § 41.

(75) **Solid Couplings.**—Fig. 137 shows the solid coupling of Whitworth's compressed steel hollow propeller shafting. One

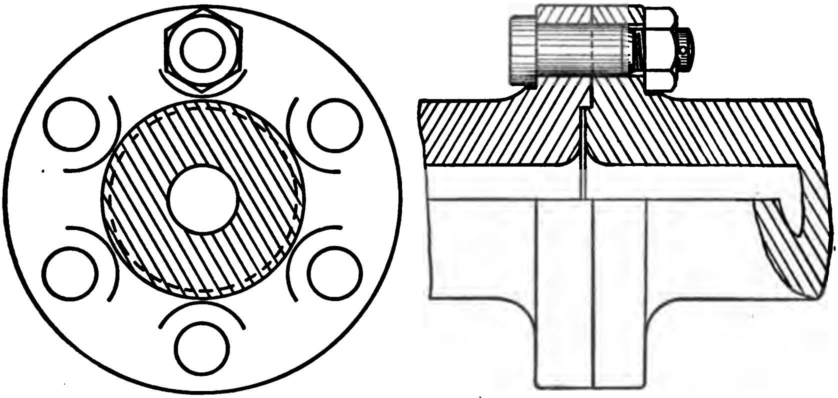


Fig. 137.

flange is made with a projecting collar, and the other with a corresponding recess into which the collar fits, but not for the whole depth. This helps to bring the shafting into line, and also makes it easier to fit in the bolts. The screw of the bolt is smaller than the plain part, otherwise it would probably be injured in driving the bolts home, as they are made a tight fit. The strength of the bolt to resist shearing is evidently to be calculated on the larger section, as they will only yield between the couplings. Notice that both the head and nut are thinner than the usual standard, since they resist no tensile stress except that due to screwing up, also that the couplings are faced up, and part of the fillet removed to give a flat surface for the bolt head and nut.

(76) **Size of Bolts for Couplings.**—The diameter and number of bolts are found as follows :—

d_b = diameter of bolts, n = number of bolts.
d_s = diameter of shaft, R = radius of bolt circle.
Resistance to shearing of bolts × R = resistance to torsion of shaft.

$$\left(d_b^2 \times \frac{\pi}{4} \times n\right) f_s \times \mathrm{R} = 0{\cdot}196\ d^3 . f$$

taking f = 9,000 lbs., and f_s = 8,000 lbs. for wrought iron.

$$d_b = 0{\cdot}53\sqrt{\frac{d^3}{n\mathrm{R}}}$$

Evidently, then, it is necessary to settle on two of these three unknowns—d_b, n, or R—before the third can be found. Generally n and R are first decided, the latter, of course, only approximately. The following will help as a guide :—

Shafts up to 3″ diameter,	$n = 4$	$d_b = \frac{3}{4}''$ or $\frac{7}{8}''$.
Shafts from 3″ to 6″ diameter,	$n = 6$	$d_b = 1''$ to $1\frac{1}{4}''$.
Shafts from 16″ to 19″ diameter,	$n = 6$ or 8	$d_b = 4''$ to $4\frac{1}{2}''$.

As a matter of fact, the bolts in shaft couplings are usually fitted of larger diameter than given by above formula, though there appears to be no reason why the extra strength is needed, unless to allow for any possible bending stress produced by the shafts running out of line.

Example.—Find diameter and number of bolts, and radius of bolt circle, for the cast-iron flange couplings of a 3″ shaft, boss $5\frac{3}{4}''$. Shaft and bolts of wrought iron.

As couplings are of cast iron, the bolts will be further from centre than in solid couplings. Say $n = 6$, and assuming $d_b = \frac{3}{4}$, then

$$\mathrm{R} = \frac{5\frac{3}{4}}{2} + \frac{3}{4} + \frac{1}{8} = 4\tfrac{1}{8} \text{ approximately;}$$

$$\text{then } d_b = 0{\cdot}53\sqrt{\frac{d^3}{n\mathrm{R}}}$$

$$= 0{\cdot}53\sqrt{\frac{3^3}{6 \times 4\frac{1}{8}}} = 0{\cdot}58''$$

hence $\frac{5}{8}''$ bolts would be ample, but, as actually fitted, there were $6 - \frac{3}{4}''$ bolts, radius of bolt circle $4\frac{1}{4}''$.

Students should notice that the effective resistance of coupling bolts increases with their distance from the centre, hence there need be no fear in increasing the bolt circle radius should that be desirable.

(77) **Drawing Shaft Couplings.**—In drawings of couplings it is usual to show a front and end elevation, the former being half in section, as in Figs. 134 to 137. The key should be drawn with its mean thickness at the centre of its length. The following is the order of drawing a flanged coupling :—

1. Draw in centre lines and shaft in both views.
2. Show boss of coupling and key.
3. Draw lines showing flanges in front elevation.
4. Decide size of bolts and radius of bolt circle.
5. Draw in bolts in both views.
6. Decide diameter of flanges and finish.

EXAMPLES.

Make working drawings of the following shaft couplings :—

EX. 5.—Box coupling (Fig. 134) for shafts of 3″ diameter, ends of shafts to be enlarged in diameter for key ways. Scale—full size.

EX. 6.—Butler's friction coupling (Fig. 135) for 3″ diameter shafts—cone sleeves, 6″ long; largest diameter, $4\frac{1}{4}$″; sleeve not to meet by $\frac{3}{4}$″; coupling, 12″ long, $6\frac{3}{4}$″ outside diameter. Scale—full size.

EX. 7.—Cast-iron flange couplings (Fig. 136) for shafts of 3″ diameter, ends of shaft not to be enlarged. Six bolts, $\frac{3}{4}$″ diameter. Scale—6″ = 1′.

EX. 8.—Solid flange coupling for hollow shafts (Fig. 137)—outside diameter, $16\frac{1}{2}$″; inside diameter, 6″, increasing to 11″ at flanges; thickness of flanges, $4\frac{1}{2}$″; six bolts $4\frac{1}{4}$″ diameter, 4″ diameter at screwed part; nuts $2\frac{3}{4}$″ thick. Scale—3″ = 1′

SECTION XXII.

RIVETED JOINTS.

(78) **Forms of Rivets.**—Rivets are the simplest permanent fastenings, and are extensively used for the connection of plates and bars in all forms of engineering and metal work construction. They are made from round bar, either of steel, iron, copper, or brass, according to the work on which they are used. Before riveting they have the form of either (*a*) or (*b*) in Fig. 138, the former being known as "*pan heads*," and the latter as "*snap heads*," and they are made to connect together two or more plates by first heating them, when made of iron or steel, to a good red heat, passing them through holes in the plates, and knocking down the heated ends which project beyond the plates to form a second head. Fig. 139 (*a*) shows a rivet in position connecting two plates before riveting, and Fig. 139 (*b*) after riveting with a snap head.

Rivet holes in plates and bars are either punched or drilled, and the riveting is done either by hand or machine, the head of

Fig. 138.

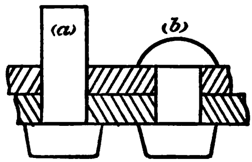


Fig. 139.

the rivet being held against a block of iron called a "*dolly*" while the end is being knocked down.

(79) **Forms of Finished Rivets.**—The usual forms of rivets after riveting is shown in Fig. 140, *a*, *b*, *c*. The ordinary pro-

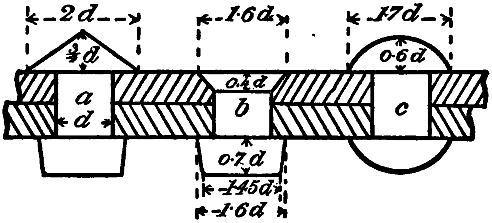


Fig. 140.

portions for the heads are marked on in terms of the diameter *d;* these differ slightly with different makers, but need not be considered by the student as unchangeable, as each maker has his own standard of sizes, and it is not usual to show the rivets on drawings of riveted work, it being quite sufficient to represent them by their centre lines, and by circles showing the rivet holes. Anything further than this would be a waste of time.

Fig. 140.—(*a*) Hand-riveting; rivet made with pan head, finished with conical head. Snap heads are also made by hand, using flogging hammers on a "snap tool."

(*b*) Hand-riveting; pan head rivet, finished with countersunk head. These are used when the surface requires to be left flush, but they are weaker than ordinary rivets.

(*c*) Machine-riveting. Snap head rivet, finished with snap head.

EXAMPLE.

EX. 1.—Make a full-sized drawing in sectional elevation and plan showing the three forms of riveting as in Fig. 140; plates, $\frac{3}{4}''$ thick; rivets, $1\frac{1}{16}$ diam.

In nearly all riveted work the rivets are arranged so that the forces acting upon them shall produce shearing stresses, and not tensile, as in the latter case the rivets are untrustworthy, the heads being liable to fly off. The following represent the ordinary arrangement of different connections of plates when riveted together, the arrows showing the direction of the forces acting upon them:—

(80) **Single and Double Riveted Lap Joints** (Fig. 141, *a*, *b*).—In these joints the plates are simply lapped one over the

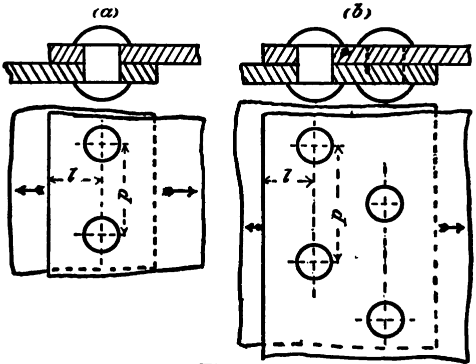


Fig. 141.

other, and riveted by either a single row of rivets (*a*) or a double row (*b*). These are the most common among riveted joints. The arrangement of rivets in Fig. 141, *b*, is termed "chain" or "zig-zag" riveting.

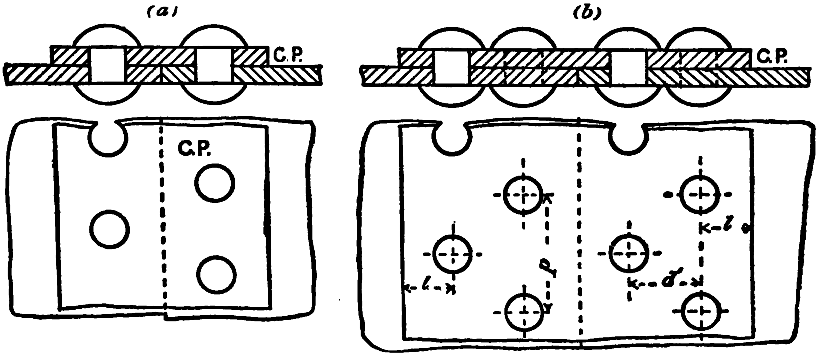


Fig. 142.

(81) **Single and Double Riveted Butt Joints** (Fig. 142 *a*, *b*).—The plates in these joints are butted together, and a narrow strip of plate, marked C.P., called a "butt strip" or "cover plate," is placed over and riveted to both plates with either a single row of rivets (*a*) or a double row (*b*).

Butt joints are also made with double cover plates, one on each side, single or double riveted.

(82) **Combined Lap and Cover Plate Joint** (Fig. 143).—This joint was first introduced in America, and is occasionally used in this country for locomotive work. There are three rows of rivets, the centre row having twice as many rivets as the outside

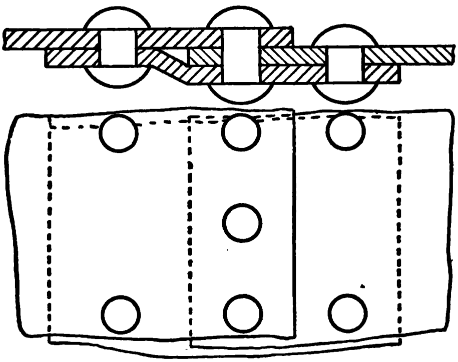


Fig. 143.

rows. The form of the cover plate makes the joint a rather expensive one.

(83) **Proportions of Riveted Joints.**—In any form of riveted joint having the plates subject to tensile and the rivets to shearing stresses, it would be a comparatively easy matter to design the joint for uniform strength, if the relation of the shearing resistance of the rivets to the tensile resistance of the plates could be exactly known, and if the strength of the plates per square inch through the line of rivet holes could be taken as equal to the strength of the solid plate, and also, if the total shearing stresses on the rivets could be assumed as equal to, and produced by, the tensile stresses in the solid plate alone.

But experiment and experience has shown that these assumptions must not be made, and hence the correct proportioning of a riveted joint is not so easy a matter. To begin with the ratio of the "rivet's shearing strength" to the "plate's tensile strength" is different when the joint is made of steel plates with iron

rivets than when it is made of steel plates with steel rivets, and obviously the proportions would also be different. Then again it is found that the tensile strength of the plates per square inch through the rivet holes is either less or greater than the strength of the solid plate, according to whether the plates are of iron or steel, and have the rivet holes punched or drilled. And lastly, a riveted joint is always tending to bend at the joint, so as to bring the two plates in the same plane, as shown in Fig. 144, and this bending action increases the stress in the joint beyond that due to the tensile forces in the solid plate.

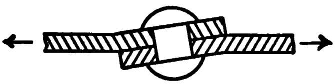


Fig. 144.

(84) **Strength of Plates for Riveted Joints.**—As a result of numerous experiments, the most reliable of which were conducted by "The Institution of Mechanical Engineers Research Committee on Riveted Joints," a series of average values of the tensile and shearing resistances at the joint can be framed, which allow for these various effects, and make it possible to design a riveted joint with considerable confidence for all the different combinations of plates and rivets. The following table of these *average* values agree very nearly with those given by Professor Unwin in his *Machine Design*, and it must be understood that the figures in column I. are the actual tensile and shearing resistances of the solid plate and rivet bar, while columns II. and III. are the nominal resistances at the joint, which differ from the original resistances owing to their allowing for the actions of bending, &c., just specified. It is, therefore, the figures in columns II. and III. which must be worked to in considering the strength at the joint:—

	I. Real Strength. Tons per sq. inch.	II. Single Riveted Nominal Strength at Joint. Tons per sq. inch.	III. Double Riveted Nominal Strength at Joint. Tons per sq. inch.
Iron plates, Drilled, .	20·5	18	19·5
,, Punched, .		16	17·5
Steel plates, Drilled, .	27·5	27·5	29·5
,, Punched, .		25	27·5
Rivet Iron, Tensile, .	27·5	...	...
,, Shearing, .	22·0	19·0 Drilled; 20·5 Punched	Same as for simple riveted joints.
Rivet Steel, Tensile, .	31·0	...	
,, Shearing, .	23·5	22·0 Drilled; 23·5 Punched	

(85) **Pitch of Riveted Joints.**—The distance between the centres of adjacent rivets is called the pitch. In the figures of the riveted joints it is marked p.

Let t = thickness of plate.
d = diameter of rivets.
p = pitch of rivets.
l = overlap of plate = distance between centre of rivet hole and edge of plate.

(86) **Relation between Diameter of Rivet and Thickness of Plate.**—There is a limit to the size of hole which can be punched in a given thickness of plate, without crushing the punch, and hence when rivet holes are punched it is necessary to keep within this limit, by arranging the diameter accordingly. But no such limit is found in drilling rivet holes, and the diameter can then be settled by considerations of uniform strength as shown later, the chief point being that, the bearing pressure of the rivet against the plate shall not exceed about 40 tons per square inch. Considerable difference of opinion exists as to the importance of this bearing pressure, and it is generally conceded that with ordinary joints designed for equal strength, the bearing pressure is sufficiently allowed for The rule which agrees most nearly with practice is the following:—

Diameter of rivet, $d = 1{\cdot}25\ \sqrt{t}$ where t = thickness of plate.

(87) **Strength of Riveted Joint.**—It is sufficient to consider the strength of a narrow strip of the joint connected by one rivet, and for convenience we will take a strip of width equal to the pitch p (Fig. 145).

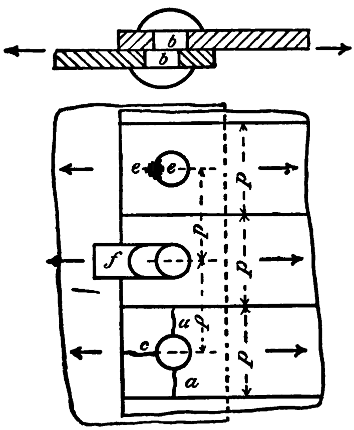


Fig. 145.

The joint may fail in one of three ways—

I. By tearing the plate through the weakest section between the rivet holes as at a.

Resistance of plate $= t\,(p - d)\,f_t$.

II. By shearing through the rivet as shown at b.

Resistance of rivet $= \left(d^2 \times \frac{\pi}{4}\right) f_s$.

III. By breaking across from the rivet hole to the outer edge of the plate, as at c. The piece of plate in front of rivet may be

regarded as a beam, of depth $l - \frac{d}{2}$ and breadth t, supported at both ends, the span being d, and loaded uniformly with a total load equal to T, the tensile stress in the strip of plate of width $(p - d)$. Then bending moment $= \frac{1}{8}$ Td, and the modulus of the section at $c = \frac{bd^2}{6} = \frac{t\ (2l - d)^2}{24}$, and therefore the BM $= \frac{t\ (2l - d)^2}{24} f$, when f = maximum tensile or compressive stress in the outer fibres, therefore $\frac{Td}{8} = \frac{t\ (2l - d)^2}{24} f$. From these two expressions we have—

$$\text{Resistance to breaking at } c\text{, when equal to } T = \tfrac{1}{3}\,\frac{t\ (2l - d)^2}{d} f_t.$$

It would also seem that the joint might fail by compression of the rivet and plate at their common bearing surface at e in Fig. 145,

$$(\text{Resistance to compression} = d\,.\,t\,.\,f_c),$$

or by shearing away the plate in front of the rivet, as at f,

$$(\text{Resistance to shearing stress} = 2\ (l \times t \times f_s)),$$

but if the joint is made strong enough to resist the action of II. and III., it is more than strong enough to resist both these actions, and hence they need not be considered.

For uniform strength the resistances of I., II., and III. must be equal, therefore—

I. and II. To find p in terms of d.

$$t\ (p - d) f_t = 0{\cdot}78\ d^2 \times f_s$$

$$\text{Substituting } \frac{d^2}{1{\cdot}56} \text{ for } t \text{ since } d = 1{\cdot}25\ \sqrt{t}.$$

$$p = d + 1{\cdot}2 \frac{f_s}{f_t}$$

Taking the values of f_s and f_t given in the table on p. 207, the following values of p are obtained:—

Iron plates, Iron rivets,	Punched holes	$p = d + 1{\cdot}5''$
	Drilled holes	$p = d + 1{\cdot}4''$
Steel plates, Steel rivets,	Punched holes	$p = d + 1{\cdot}13''$
	Drilled holes	$p = d + 1{\cdot}0''$

II. and III. To find overlap l in terms of d.

$$\tfrac{1}{3}\ \frac{t\ (2l - d)^2}{d} f_t = 0{\cdot}78\ d^2 \cdot f_s$$

$$\text{Substituting } \frac{d^2}{1{\cdot}56} \text{ for } t \text{ as before.}$$

$$l = 0{\cdot}95 \sqrt{\frac{f_s}{f_t}}\, d + \frac{d}{2}$$

The largest values for $\frac{f_s}{f_t}$ is 1·3 for iron plates and iron rivets, punched holes, and substituting this:—$l = 1{\cdot}1\sqrt{d} + \frac{d}{2}$. This gives the following rule:—

Practical Rule for Overlap—

$l = 1{\cdot}7\,d$ for rivets up to $\frac{7}{8}''$ diameter.
$l = 1{\cdot}5\,d$ for rivets over $\frac{7}{8}''$ „

(88) Pitch of Single Riveted Joints.

Thickness of Plate. t.	Diameter of Rivet. d.	Iron Plates and Iron Rivets.		Steel Plates and Steel Rivets.	
		Punched Holes.	Drilled Holes.	Punched Holes.	Drilled Holes.
	$1{\cdot}25\sqrt{t}$	$d + 1{\cdot}5''$	$d + 1{\cdot}4''$	$d + 1{\cdot}13''$	$d + 1{\cdot}0''$
$\frac{1}{2}''$	$\frac{7}{8}''$	$2\frac{3}{8}''$	$2\frac{1}{4}''$	$2''$	$1\frac{7}{8}''$
$\frac{3}{4}''$	$1\frac{1}{16}''$	$2\frac{9}{16}''$	$2\frac{1}{2}''$	$2\frac{3}{16}''$	$2\frac{1}{16}''$
$1''$	$1\frac{1}{4}''$	$2\frac{3}{4}''$	$2\frac{5}{8}''$	$2\frac{3}{8}''$	$2\frac{1}{4}''$

(89) **Double-riveted Joints.**—In a double-riveted joint the plate may tear *diagonally* between the holes, as well as between the holes parallel to the outer edge of the plate, and, therefore, the diagonal distance must be proportioned for equal strength. Experiment has shown that the strength of a plate torn diagonally is only about three-fourths the strength when torn parallel, so that the distance between the two rows of rivets is made sufficient to allow for this reduction.

The rule adopted in practice is as follows:—

Distance between rows of rivets in double-riveted joints equals twice diameter of rivets (marked d' in Fig. 142b).

(90) Pitch of Double-riveted Joints.

Thickness of Plate. t.	Diameter of Rivet. d.	Iron Plates and Iron Rivets.		Steel Plates and Steel Rivets.	
		Punched Holes.	Drilled Holes.	Punched Holes.	Drilled Holes.
$\frac{1}{2}''$	$\frac{7}{8}''$	$4''$	$3\frac{1}{2}''$	$3\frac{1}{8}''$	$3''$
$\frac{3}{4}''$	$1\frac{1}{16}''$	$4\frac{1}{4}''$	$3\frac{5}{8}''$	$3\frac{1}{4}''$	$3\frac{1}{8}''$
$1''$	$1\frac{1}{4}''$	$4\frac{1}{2}''$	$4''$	$3\frac{1}{2}''$	$3\frac{1}{4}''$

(91) Thickness of Butt Straps.—

Thickness of single strap = $1\frac{1}{8}\ t$.
Thickness of each double strap = $\frac{5}{8}\ t$.

(92) **Lozenge Joint.**—Fig. 146 shows the riveted connection of two plates of a girder for roof or bridge work, and known as a lozenge joint. The plates are either butted together and riveted, as in the figure, to a single or double butt strap; or the plates are simply lapped one over the other. By arranging the rivets

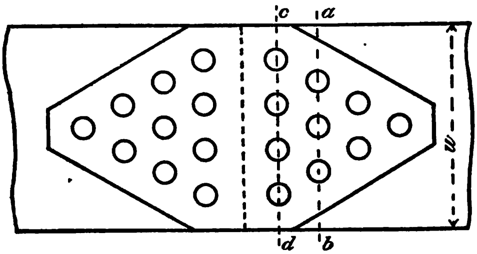


Fig. 146.

in lozenge order the joint can be made of nearly uniform strength throughout, as, for example, before the joint can fail across the line $a\,b$, three rivets must be sheared, and the plate must tear through a section, $t(w - 3d)$;

$$\text{Resistance} = 3\left(d^2 \times \frac{\pi}{4}\right) f_s + t(w - 3d) f_t.$$

And similarly, before it can fail across the line $c\,d$, six rivets must be sheared, and the plate must tear through a section, $t(w-4d)$;

$$\text{Resistance} = 6\left(d^2 \times \frac{\pi}{4}\right) f_s + t(w - 4d) f_t.$$

Hence, as the section of the plate decreases, the number of rivets to be sheared increases.

EXAMPLES.

Make drawings of the following riveted joints to a scale of $6'' = 1'$. Show two views, a sectional elevation, and a plan:—

EX. 2.—Single and double-riveted lap joints—$\frac{3}{4}''$ plates—iron plates and rivets, with drilled holes (Fig. 141).

EX. 3.—Single-riveted butt joint with two cover plates—

plates 1″ thick—steel plates and rivets, drilled holes (see Fig. 142*a*).

EX. 4.—Combined lap and cover-plate joint (Fig. 143)—plates $\frac{3}{4}$″ thick—steel plates and rivets, drilled holes.

(Centre row of rivets ordinary pitch, outer rows as close in as possible for convenient riveting.)

EX. 5.—Lozenge joint (Fig. 146)—plates 1″ thick—lapped one over the other without cover plate. Width of plate 15″.

(Shearing strength of rivets to equal tensile strength of plate at weakest section—*i.e.*, through first or second row of rivet holes.)

(93) **Efficiency of Riveted Joints.**—It is evident that a riveted joint cannot possibly be as strong to resist tensile forces as the plate of which it is made, simply because the section through the rivet holes is always less than the section of the solid plate. The ratio between the strength of the joint and of the plate is termed the *efficiency of the joint.* If the tensile resistance per square inch was the same through the holes as through the solid plate, the efficiency would evidently be $\frac{t(p-d)f_t}{t \cdot p \cdot f_t} = \frac{p-d}{p}$, but as this is not the case, for reasons already seen, the efficiency is expressed by $K \cdot \frac{p-d}{p}$ where K = ratio of the resistance per square inch through the holes, to the resistance per square inch through the solid plate, and varies with different joints. The "theoretical efficiency" $\frac{p-d}{p}$ varies from 75 per cent. to 100 per cent., and the "actual efficiency" $K \cdot \frac{p-d}{p}$, from 41 per cent. to 72 per cent.

The student will see the importance of knowing the efficiency of the joint in designing riveted structures, as the strength of the structure cannot evidently be greater than the strength of its weakest joint. Hence, in designing a boiler of plates having a tensile resistance per square inch equal to T, the working stress must be proportioned to E T where E = efficiency of joint, since this is the actual strength of the plate.

(94) **Working Stress.**—In ordinary boiler and bridge work the working stress may be one-fourth to one-fifth of the breaking stress (Unwin). A high factor of safety is necessary to allow for loss due to corrosion

(95) **Connection of more than two Plates.**—It is necessary in boiler work to rivet joints where three or four plates join, as in the case where the longitudinal and circumferential joints meet. An example of the connection of four plates is shown in Fig. 147, the plates being marked A, B, C, D. The plates B and D have their corners thinned out as shown, so as to lap one over the other, and not exceed the thickness of one plate, each thin corner being lengthened to be held by two rivets.

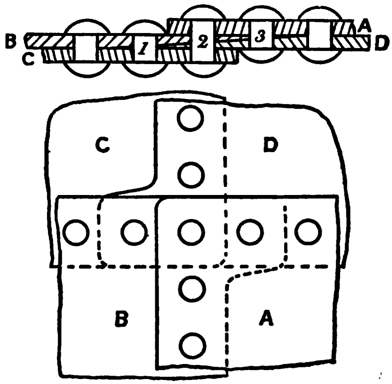


Fig. 147.

Thus the three rivets, 1, 2, 3, pass through three plates.

(96) **Rolled Bars.**—In Fig. 148, *a*, *b*, *c*, *d*, are shown sections of bars commonly used for different connections of plates. They are rolled in long lengths of various standard sizes, in either iron or steel. The width of the flanges increase by $\frac{1}{8}''$ for angle bars (*a*), and by $\frac{1}{4}''$ for other bars. Thickness of flanges increase by $\frac{1}{16}''$ for angle and tee bars of small size, for larger sizes and for other bars, by $\frac{1}{8}''$. The thickness at the root is slightly greater than at the edges. No sections should be shown on a drawing which cannot be found in the makers' lists.

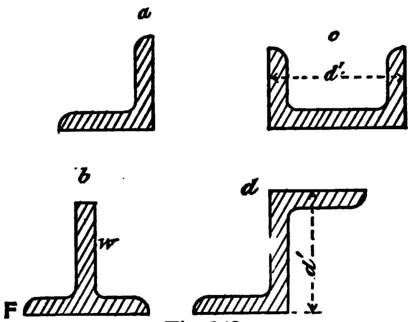


Fig. 148.

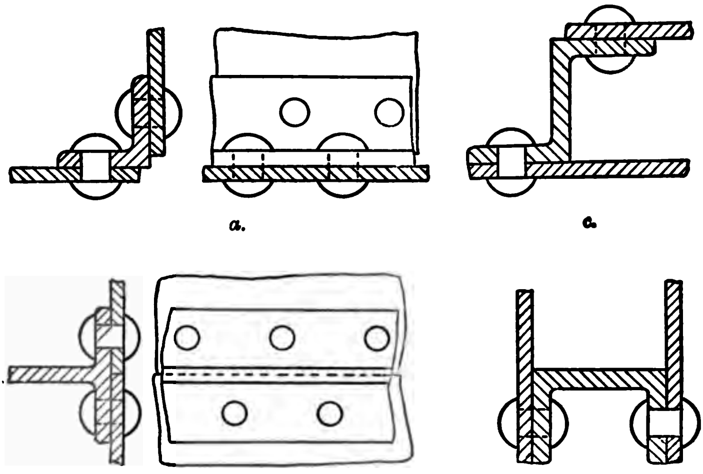
(*a*) *Angle Bars.*—Made with flanges of equal or unequal width. Sizes, from $\frac{3}{4}'' \times \frac{3}{4}'' \times \frac{1}{8}''$ to $6'' \times 6'' \times \frac{3}{4}''$. Weight, from 0·89 lb. to 28·7 lbs. per foot of length.

(*b*) *Tee Bars.*—Made with the flange F and web W of equal or unequal width. Sizes, from $3'' \times 3'' \times \frac{5}{16}''$ to $7'' \times 10'' \times \frac{3}{4}''$ (the first size is width of flange). Weight, from 7·2 lbs. to 45 lbs. per foot of length.

(*c*) *Zed Bars.*—Made with equal or unequal flanges, and of various depths. Sizes, from $3'' \times 3'' \times 2\frac{1}{2}'' \times \frac{3}{8}''$ to $10'' \times 3\frac{1}{2}'' \times 3\frac{1}{2}'' \times \frac{3}{8}''$ (the first size is the depth d^1). Weight, from 10 lbs. to 44 lbs. per foot of length.

(*d*) *Channel Bars.*—Made with flanges of equal size, and of different depths. Sizes, from $2\frac{1}{4}'' \times 1\frac{1}{4}'' \times \frac{1}{4}''$ to $12'' \times 4'' \times \frac{1}{2}''$ (the first size is the depth d^1). Weight, from 4 lbs. to 33 lbs. per foot of length.

(97) **Connection of Plates and Bars.**—The use of the above



a. *c.*

b. Fig. 149. *d.*

bars for different connections of plates in boiler and tank work is shown in Fig. 149, *a*, *b*, *c*, *d*.

(*a*) Angle bars, used to connect plates at right angles. Notice that the rivets in one flange come between the rivets in the other flange, for convenience in riveting up.

(*b*) Tee bars, used to connect plates in the same plane, and also of great service in stiffening the plates.

(*c*) and (*d*) Zed and Channel bars, used to connect plates parallel to each other, and commonly employed to connect the inside fire box to the shell plates of a locomotive type of boiler. Notice that the use of the Zed iron gives one inside joint, which is avoided in the Channel iron. The rivets in the Channel iron should be arranged as with angle bars.

(98) **Flanging of Plates.**—The simplest connection of plates

at right angles is shown in Fig. 150, where one of the plates is bent to form a flange, which is then riveted to the other plate. The inside radius of the flanged plate should not be less than four times the plate thickness. This is the common construction in boiler work, where the flanged plate is the flat end plate, riveted to the circumferential shell plate. It possesses the great advantage over the angle bar, Fig. 149, *a*, as giving only one riveted joint.

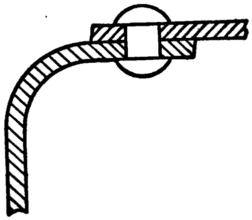


Fig. 150.

EXAMPLES.

Make working dimensioned drawings, half full size, showing two views of the following connections of plates and bars :—

EX. 6.—Connection of four iron plates $\frac{3}{4}''$ thick, as in Fig. 147, single riveted iron plates, drilled holes.

EX. 7.—Connections of plates and bars as follows, all iron, with iron rivets and drilled holes :—

(*a*) Angle iron, .	$3'' \times 3'' \times \frac{1}{2}''$,	plates $\frac{1}{2}''$,	rivets $\frac{7}{8}''$	diam.	Fig. 149,	*a*.
(*b*) Tee iron, . .	$4'' \times 3'' \times \frac{1}{2}''$,	,,	,,	,,	,,	*b*.
(*c*) Zed iron, . .	$4'' \times 3'' \times 3'' \times \frac{1}{2}''$,	,,	,,	,,	,,	*c*.
(*d*) Channel iron,	$4'' \times 3'' \times \frac{1}{2}''$,	,,	,,	,,	,,	*d*.

SECTION XXIII.

LUBRICATED BEARINGS AND SHAFT PEDESTALS.

THE design of lubricated bearings is a subject of great importance, affecting as it does the efficient working of nearly all classes of machinery. It is far too large to permit of any but general principles being explained here, the student being referred for fuller information to the chapter on "Journals" in Professor Unwin's *Machine Design*, and to the proceedings of the Institution of Mechanical Engineers' Research Committee on "Friction."

All rotating shafts and axles of machines require supporting in such a way that as little as possible of the power they transmit is lost. Such supports have received the general name of "*bearings*," common examples of which will be familiar to engineering students in their application to shafting in workshops, spindles of lathes, and to the crank shafts of engines.

(99) **Journals.**—That part of the shaft or axle supported by the bearing is known as the "*journal.*" Evidently its diameter and length must be proportioned according to the forces acting upon it and to the power it transmits. Such considerations of strength are, however, too advanced for the limits of this book. But apart from the question of strength a journal must be proportioned according to the pressure it exerts upon the bearing which supports it, and to its other working conditions.

(100) **Friction of Journals.**—One effect of the pressure exerted by a journal upon its bearing is to produce a frictional resistance to movement. Part of the work which the journal is transmitting must be expended in overcoming this resistance, which work is converted into heat, and raises the temperature of the rubbing surfaces. If this increase of temperature continues, the journal will seize in the bearing, producing more or less serious results. The object of lubrication is to reduce this frictional resistance to a minimum, thereby ensuring a good working bearing and making the useful work performed by the shaft as large as possible.

Generally speaking, the friction of rubbing surfaces may be regarded as follows:—Let W = the pressure between the surfaces producing the friction. F = the force required to overcome the friction, and, therefore, a measure of the "*frictional resistance.*" Then, in general terms, $F = \mu W$, where μ is a certain factor known in mechanics as the "*coefficient of friction,*" but which for journals is not necessarily a simple factor. Evidently then, with a given value of the load W, the smaller the coefficient of friction μ, the less is the frictional resistance F. The aim in lubricated bearings is to keep μ as small as possible, and, therefore, to reduce the frictional resistance F to a minimum.

The frictional resistance of lubricated bearings may be stated in general terms to depend upon the following conditions:—

I. The pressure per square inch between the sliding surfaces.
II. The velocity of the sliding surfaces (ft. per min.)
III. The materials of which the sliding surfaces are composed.
VI. The kind of lubricant used, and the method of lubrication.

(101) I. **Pressure between the Sliding Surfaces.**—The effect of increasing the pressure between a journal and bearing is that the lubricant is squeezed out from between the sliding surfaces, and the journal heats and seizes. Evidently, then, the safe working pressure depends entirely upon the manner of lubrication, and hence a bearing which may seize with one method of lubrication will run quite smoothly with another method. For bath lubrication, where part of the journal runs

continually in a bath of oil, the frictional resistance has been found by experiment to be practically constant with different pressures, a result the very opposite to what occurs with unlubricated or badly lubricated surfaces.

The limit of working pressure also depends upon the manner in which the pressure acts upon the bearings; in ordinary workshop shafting the pressure is practically always upon the same part of the bearing, and the motion is in one direction only; on engine crank pins the direction of the pressure is constantly changing, and acts alternately on the front and back halves of the bearing, while on the crosshead journals the direction of both pressure and motion changes with every revolution of the engine. In all these cases the safe working pressure has a different value.

When the diameter of the journal is d inches and length l inches, the effective bearing area of the journal is $d \times l$ square inches. Therefore, when the total load upon the journal is W lbs., the pressure p in lbs. per square inch is equal to $\frac{W}{d \times l}$.

(102) II. **Velocity of Sliding Surfaces.**—At velocities of below 100 feet per minute with high loads, the frictional resistance is generally abnormally high, but at greater velocities it has been found to vary directly with the square root of the velocity, with good lubrication, and with pressures below a certain limit. It is always assumed that the frictional resistance increases as the velocity increases, and hence bearings for journals running at high velocities are subjected to a less pressure per square inch than bearings for low velocities.

If a shaft of diameter d inches makes N revolutions per minute, then the velocity of sliding in feet per minute is equal to $\frac{\pi d N}{12}$.

(103) III. **Material of Sliding Surfaces.**—Different metals in contact give different frictional resistances, hence it is desirable to choose such materials as shall be known to run easily. For a material having suitable wearing qualities, gunmetal (a mixture of copper and tin) is found to give exceedingly good results when used in contact with iron or steel journals, it is, therefore, chiefly used as linings for all kinds of bearings, the linings being known technically as "*brasses*" or "*steps.*" Other metals such as "white metal" (a soft mixture of lead or tin, with copper and antimony), "Babbit's metal," and "Magnolia" are known to largely reduce the frictional resistances, and are, therefore, used to line the brasses of bearings, especially when of large size.

(104) **IV. Lubricants and Lubrications.**—Very little need be said here of the different lubricants; they are used either in a fluid form, as the mineral and vegetable oils, or in the more solid form of tallow, and their special qualities is more a question for the machine user than the designer. This, however, is not the case with the method of applying the lubricant, upon which indeed the whole success of a bearing depends.

(105) **Seat of Pressure.**—In all bearings there are one or more parts where a greater pressure is exerted than upon any other part, which parts may be called the "*seat of pressure.*" For example, in the main bearing of an engine crank shaft, fitted with a driving flywheel, forces act upon the bearing caused by (i.) the pressure on the crank pin; (ii.) the mass of the flywheel; (iii.) the pull of the driving belt; and if the magnitude and direction of these are known, it is easy to approximately determine the direction of the resultant pressure, which resultant will pass through the "seat of pressure." The lubricant must then enter into the brass at the opposite side of the bearing, as only by so doing can it possibly be carried round by the journal and form the necessary film between it and the brass at the part where the pressure is greatest. Hence all oil holes or channels should lead to that part of the brass where the pressure of the journal is least, and which may be called the "*off side*" of the brass.

(106) **Bath Lubrication.**—The best method of applying a lubricant is undoubtedly that known as "bath lubrication," where the journal dips in a bath of oil, some of which it carries round with it every revolution. But such a fitting is costly and complex, and is very seldom practically used. An example of its application to the bearing of an ordinary driving shaft is given in Fig. 160.

(107) **Pad Lubrication.**—The next best results are obtained by using "pads" saturated with the lubricant. As commonly fitted, it is usual to cut away portions of the step where the pressure is least, and fill in the space with a pad of soft felt, to which the oil way leads. Such an arrangement is shown in Fig. 151, where the right- and left-hand quarters of the brasses inside the flanges are cut away, the space being filled with felt, F F. The felt is kept saturated with oil from the oil cup C,

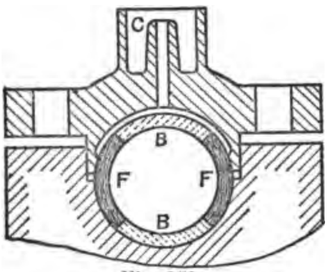


Fig. 151.

through the holes and grooves shown. Locomotive axle boxes are lubricated in this way.

(108) **Needle and Siphon Lubrication.**—The common but less efficient method is to constantly supply a small stream, or continuous drops of the lubricant to one or more parts of the journal by means of the "needle" or "siphon" lubricator.

The needle lubricator (Fig. 152) is largely used in ordinary shop shafting, and so far as it goes, is very efficient. It simply consists of a glass bottle, G, fitted with a wooden plug, W,

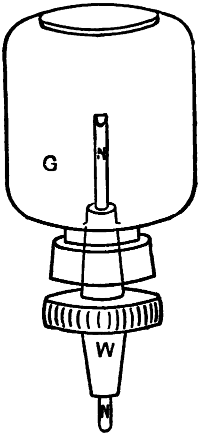


Fig. 152.

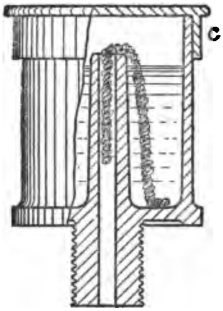


Fig. 153.

through which passes a short length of thin wire, N, the lower end of which rests upon the journal. When in good order, the very small vibrations of the journal as it rotates are supposed to cause a slow descent of the oil, which descent stops when the journal ceases to rotate.

Siphon lubricators, Fig. 153, are most commonly employed in all classes of bearings. They consist of an oil cup fitted with a cover or cap, C, and having a small tube leading from near the top of the cup to the journal, a piece of cotton wick is pushed down the tube for a short distance, the other end lying in the oil, the result being a continuous slow siphoning of the oil into the pipe through which it reaches the journal. Inside diameter of tube, $\frac{3}{16}''$ to $\frac{5}{16}''$.

The relative values of the three methods of lubrication—"bath," "pad," and "siphon," may be taken as equal to the numbers 10, 3, 2, the "bath" being the best.

Solid lubricants are best filled into a box having a screwed top, the periodical screwing down of which forces a small portion of the lubricant on to the journal.

(109) **Practical Limits of Pressure.**—The following values are those usually adhered to in practice for bearings of ordinary construction with siphon lubrication. It will be noticed that the pressure is least for those journals running at high velocities, and also that the range in some of the cases is considerable. This latter is, of course, to allow for the different speeds at which different shafting and engines run, and hence the student should select the lower values for high speeds (300 to 400 revs. per min.), and the higher values for low speeds (120 to 200 revs. per min.)

As a general rule, machine and engine makers aim at long journals and low pressures, and a consequent smooth-running bearing, especially with high speeds. Thus while the length of ordinary shafting and engine journals varies from d to $1\frac{1}{2}\,d$, where d = diameter, the length for quick-running engines and dynamos is often equal to $2\frac{1}{2}\,d$ or $3\,d$. On the other hand, when limitations of space prevent a long journal being used, close attention must be paid to efficient and continued lubrication.

Journals for—		Press. in lbs. per sq. in.
Workshop shafting,		200
Flywheel shafts,		150 to 250
Engine crank pins,	Small engines,	500 to 800
Engine crosshead pins,	Small engines,	700 to 1800
Engine girder blocks,	Small engines,	25 to 50

With bath lubrication at the highest velocities, the pressure should not exceed 500 lbs. per square inch.

(110) **Pedestal Bearings or Plummer Blocks.**—An example of the simplest make of adjustable bearing or plummer block is shown in Figs. 154 *a* and *b*. It represents the construction commonly adopted by makers of mill and workshop shafting for ordinary driving purposes, and may be bolted to hanging brackets or wall boxes. In general principles it is representative of larger pedestals for the more important work of crank shaft and flywheel bearings, to which the following remarks may be taken to generally apply:—

The pedestal P is a cast-iron block provided with a larger base plate, P B, for rigidly bolting the pedestal down, and a loose cap C, held to the pedestal by bolts *b b*, which in Fig. 154 pass right through from the base, but which in larger sizes are screwed or cottered into the pedestal. Loose brasses, B B, fit between the pedestal and the cap, and form the bearing surface for the

journal. Lubrication is effected in the block of Fig. 154 by fitting a glass lubricator in the hole shown in the top of the cap,

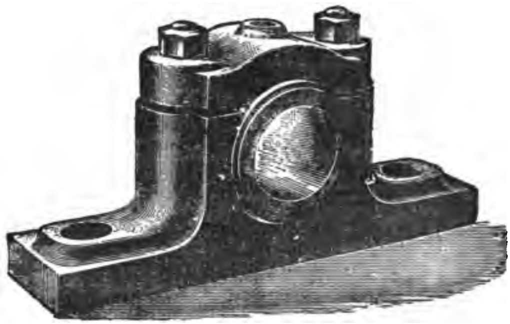


Fig. 154*a*.

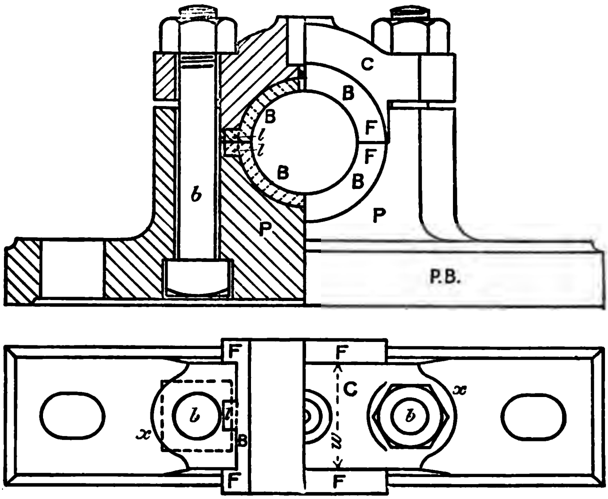


Fig. 154*b*.

and in larger sizes by a separate siphon lubricator, or a lubricator cast with the block.

(111) **Brasses or Steps.**—These consist of two half bushes or sleeves of gunmetal, B B, accurately bored to surround the journal, which fit into the cast iron pedestal P, and are kept from turning by one of the following devices:—(1) When the

pedestal is bored with a cylindrical hole, the brasses may be fitted with small rectangular lugs, *l*, which fit into recesses in the pedestal and cap, as seen in Fig. 154*b*, or they may have a small length of round rod driven into the top and bottom of the brasses, and fitting into holes drilled in the pedestal and cap; these are both cheap stops, but are not adopted in larger pedestals. (2) The pedestal may be finished with a square or a hexagonal hole, and the brasses cast square or hexagonal for a certain part of their length, about $\frac{1}{2}''$ to $\frac{3}{4}''$, as shown in Figs. 155*a* and *b*. One of these is generally adopted for pedestals of large size, the former being the cheaper.

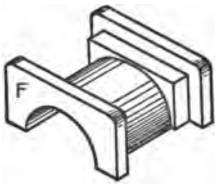


Fig. 155*a*.

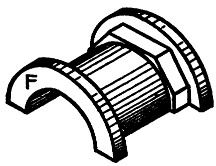


Fig. 155*b*.

The brasses are prevented from moving along the shaft by side flanges, F F, between which the pedestal and cap fit. These are shown on the brasses of Figs. 154 and 155. The brasses cannot then, of course, be pushed into place from the side of the pedestal, but are dropped in from the top, and should fit without play.

Brasses should be divided across the line of least pressure at which part they may be slightly thinner than at the parts where the pressure is greatest and where they will wear most.

(112) **Adjustment of Brasses.**—In the majority of pedestals it is only necessary to allow of a vertical adjustment, the wear only taking place at the bottom of the lower brass. This adjustment is effected by filing off a part of one or both brasses from the surfaces where they meet, and screwing down the cap C; to allow of this the cap should be shown clear of the pedestal by a distance of from $\frac{3}{16}''$ to $\frac{1}{2}''$, depending upon the size, and should only bear on the top brass. In engine and flywheel bearings the brasses are separated by packing strips of brass which, by being made thinner, enable the wear to be taken up.

Pedestals for engine crank shafts require horizontal adjustment, as the pull and thrust along the connecting-rod produces side wear; a convenient arrangement for effecting this is shown in Fig. 158. When fitted with a heavy flywheel or driving-

wheel they may also require vertical adjustment, and the brasses are then made in four sections, and are arranged as in Fig. 156, which represents the brasses (marked B) for the pedestal of an engine crank shaft fitted with a heavy driving-wheel, the pull on which was nearly horizontal. In this pedestal a vertical adjustment is obtained by screwing down the cap C, and a horizontal adjustment by means of the wedge blocks W W, which are moved by the screws S S.

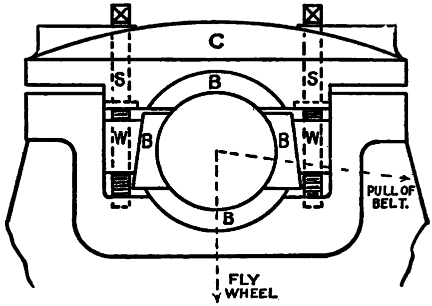


Fig. 156.

(113) **Pedestal and Cap.**—The width of the pedestal and cap (marked *w* in the plan of Fig. 154*b*) is decided by the length of the brasses inside the flanges, but as a rule it is unnecessary to adopt this width throughout, and it is made less round the cap bolts, as shown. To a certain extent the length of the base depends upon the distance apart of the cap bolts, but it is not advisable to cut down the metal of a pedestal to the smallest amount, as it would probably then lack stiffness and rigidity, and would not give steady running, hence the student should not look for a statement of the exact proportions to be observed since a pedestal is not designed by considering the stresses acting upon it, to resist which it is probably abnormally strong, but rather with the idea of properly supporting the brasses with sufficient strength and stiffness.

In pedestals where the pressure acts horizontally upon one or both sides of the brasses, as in crank pin bearings or where the pull of the strap on the driving-wheel is horizontal, the pressure tends to turn the pedestal about one end of the base and produces a tensile stress upon the bolt holding down the other end of the base; this turning effort increases with the height of the centre of the journal above the base, and it is, therefore, *desirable to keep the brasses low in the pedestal.* The examples given will show that this is generally done.

Notice that in order to reduce the amount of machinery and surfacing to a minimum, the brasses are only fitted for a short distance inside the flanges, the other parts of the pedestal and cap and brasses being left as cast, and clear of each other (see Figs. 155*a* and *b*). Also, that the pedestal base only rests upon

a narrow strip round the edge (known as "chipping strips"), that bosses are generally cast for all nuts to bed upon, and that a narrow projecting strip of metal is also cast around the pedestal and cap where the flanges of the brasses fit. The holes in the pedestal base are made long to allow for adjustment.

(114) **Bolts in Pedestal Caps.**—These are invariably pitched as close together as possible, and frequently cut into the sides of the brasses for bearings, having only vertical wear. When studs are used there is, however, a limit to their nearness to the brass for the reasons given in § 42, and hence bolts are frequently used as in Fig. 154, which pass right through the pedestal from bottom to top. But with angle pedestals, as in Fig. 157, the use of bolts is not possible, and then the custom is to drive small cotters through the pedestal and bolts, and thus gain a compact design.

A pedestal should never be arranged so that much, if any, pressure comes upon the cap, hence the cap bolts are not required to stand much stress. The metal around the bolts must be sufficient to extend about $\frac{1}{8}''$ beyond corners of nuts. It is better to take the centre for the arc x in Fig. 154*b*, from a point within the bolt centre, and not at the bolt centre.

(115) **Proportions of Pedestals.**—The following proportions will assist in guiding the design, d = diameter of journal :—

Length of brasses = length of journal = $1{\cdot}5\,d$ for ordinary speeds, $2\,d$ for quick speeds.

Thickness of brasses at seat of pressure = $\frac{1}{8}\,d + \frac{1}{16}''$ (minimum of $\frac{1}{4}''$). The thickness at other parts for journals above $2''$ diameter may be three-quarters of this thickness.

Flanges on brasses not less than $\frac{1}{4}''$ thick, usually $\frac{3}{8}''$ to $\frac{3}{4}''$. Flanges need not lap over the pedestal more than $\frac{3}{8}''$ to $\frac{3}{4}''$

Diameter of bolts in cap = $\frac{1}{4}\,d$ to $0{\cdot}3\,d$.

Thickness of cap = $1{\cdot}3$ to $1{\cdot}5$ times diameter of bolts.

Diameter of holding down bolts = diameter of bolts in cap + $\frac{1}{8}''$.

Thickness of base = $1{\cdot}5$ to $1{\cdot}7$ times diameter of bolts. Centres of base bolts arranged to give room for nuts to turn round without cutting into the fillet joining the pedestal to the base; metal outside holes to allow for overlapping nuts as in ordinary flanges.

(116) **Crank-shaft Pedestals.** — The "angle" pedestal in Fig. 157 is of a type now extensively used for the crank-shaft bearings of small-sized stationary engines, and is frequently cast with the engine bed—the usual angle of inclination being 45°. By arranging the cap in this way it is possible to take up the brasses, and allow for wear better than with a horizontal cap, and it thereby combines a partial horizontal adjustment without any special fittings. The cap bolts are held to the pedestal by cotters, C, driven through the bolts and the surrounding metal,

as shown in the figure for one bolt. The bolts are often made square for the length within the pedestal, to allow for the metal removed by the cotter way, and the cotters may be driven from a front face of the pedestal instead of the sides. Bolts rather larger than the usual size, and provided with lock nuts, should be fitted, as part of the pressure on the upper brass comes on the bolts. The siphon oil cup S C is cast with the cap, and should be fitted with a loose cover.

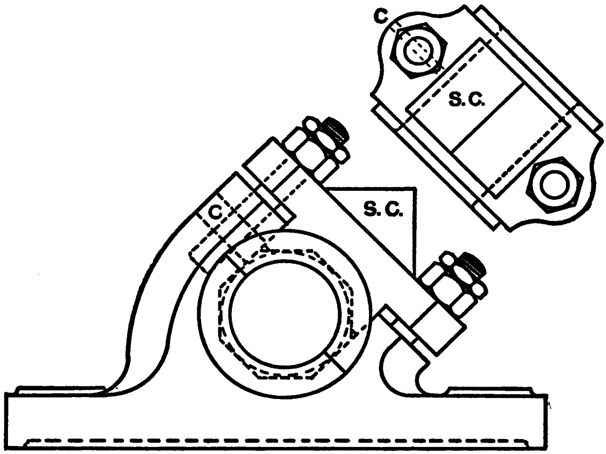


Fig. 157.

The crank-shaft pedestal of Fig. 158 is shown as fitted to the horizontal engine of Fig. 179, and is arranged for horizontal adjustment. The brasses are split in two places in vertical planes with a packing strip of leather between them, but as will be seen in the figure at A and B, are so divided, that when the packing pieces P are removed, the two halves of the brasses A and B can be turned round and lifted from the pedestal. The flanges on the brasses must be large enough to extend over the packing pieces and bear against the metal of the pedestal.

Cast-iron distance pieces, P P, are inserted between the sides of the brasses and the pedestals, which are forced against the brasses by screwing in the set screws S S, thus adjusting for wear. An isometric projection of the packing piece is shown separately at P′. Notice the recess R in the pedestal where the brasses rest, to avoid machining that part all over. The cap is held by studs screwed into the pedestal, and is provided with a siphon lubricator. cast with the cap.

It is important to remember that crank-shaft pedestals are subject to a changing stress, the pressure being first on one brass and then on the other brass. The brasses must, therefore, be well surrounded by the pedestal, so that the stresses may not come upon the cap or bolts. By referring to Fig. 154, where the pressure is practically a downward one, it will be seen that the top brass is almost completely surrounded by the cap, whereas in Figs. 157 and 158 the cap is practically a flat plate, and both the top and bottom brasses are surrounded by the pedestal, thereby ensuring considerable strength and rigidity. This difference must be borne in mind when designing. In Figs. 154

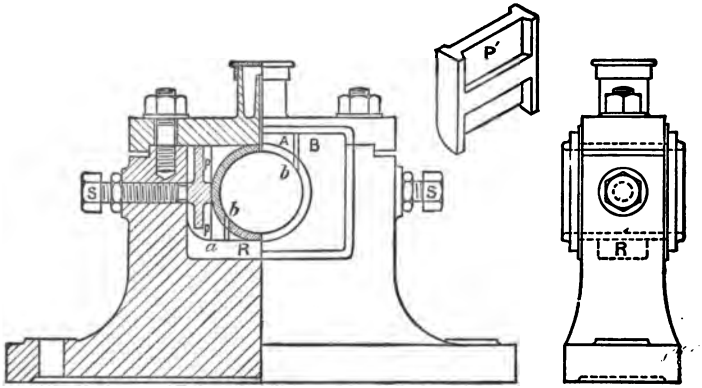


Fig. 158.

and 157 the cap is prevented from moving independently of the bolts by inside projections fitting within the pedestal, and in Fig. 158 by the outside projections which fit into outside recesses in the pedestal. Some provision of this kind must always be made.

(117) **Order of Drawing.**—The following order should be adopted in designing and drawing a pedestal. Usually three views should be shown—a front elevation half in section, an end elevation partly or wholly in section, and a plan :—

1. Decide ratio of length to diameter of brasses, and thickness; also height from base to centre, and draw brasses in all views.
2. Show thickness and position of cap, draw in bolts and proportion the metal around the bolts, and the length of the cap, allowing for overlap beyond the nut.
3. Arrange size and position of holding-down bolts in base, and length of base.

4. Complete details, not forgetting bosses, chipping strips, and arrangements for lubrication.

EXAMPLES.

Make working dimensioned drawings, showing three views of the following pedestals :—

EX. 1. — Simple pedestal for $1\frac{1}{2}''$ shaft, as in Fig. 154. Journal $2\frac{1}{4}''$ long, brasses $\frac{1}{4}''$ thick, flanges $2\frac{3}{8}''$ diam., $\frac{3}{8}''$ thick. Cap $\frac{3}{4}''$ thick, $\frac{1}{2}''$ bolts. Base $1''$ thick, $2''$ wide, $\frac{5}{8}''$ bolts. Hole in cover for glass lubricator $\frac{1}{2}''$ diam., $\frac{3}{4}''$ deep. Full size.

(Brasses divided across horizontal centre line, thickness may be uniform as size is small. Cap bolts, square heads; clearing holes, within about $\frac{1}{8}''$ of lugs on brasses. Cap is thicker next brass than where bolts pass through. Pedestals of such small size do not usually have bosses and projections for bolts and flanges of brasses, but must have under side of base recessed. Show a good fillet between pedestal and base. Arrange height so that flange of brasses is about $\frac{3}{8}''$ clear of base.)

EX. 2.—Crank-shaft angle pedestal, as in Fig. 157, for $4''$ journal, $6''$ long. Angle 45°, cap bolts $1\frac{1}{8}''$ diameter, held by cotters and fitted with lock nuts. Brasses, as in Fig. 155*b*, with circular flanges $\frac{1}{2}''$ thick. Oil box cast with cover. Full size.

(Brasses divided across 45° line, draw in brasses from proportions in text, next draw in bolts; if cotters are driven as shown in Fig. 157, then metal on each side of hole not less than bolt diameter; if driven from face at right angles to position of Fig. 157, then bolts may be closer in. Arrange metal around bolts, and length of cap. Let distance from centre of shaft to base be as small as convenient to get in base bolts on right-hand side clear of cap, so as to allow for nut. Notice that the further the base from the centre, the shorter need the base be to allow of this clearance. Provide projecting face for flanges of brasses, and bosses for base bolts.)

EX. 3.—Crank-shaft pedestal with horizontal adjustment as in Fig. 158, journal $3\frac{1}{2}''$ diameter, $5''$ long. Height from base to centre of shaft $8''$. Brasses as in Fig. 155*a*, to overlap pedestal on each side $\frac{5}{8}''$, flanges $\frac{3}{8}''$ thick. Base $3\frac{1}{2}''$ wide. Round cup for lubricator, cast with cap, $2\frac{1}{4}''$ deep, $2''$ diam. outside. Set screws for adjustment $\frac{7}{8}''$ diam. with lock nuts. Bosses for holding down bolts and for set screws. Scale, $6'' = 1'$.

(Draw in brasses, which may be reduced in thickness at top and bottom by planing off a part in machining up the squared parts next to flanges. Brasses split as shown in vertical planes, and so that a line joining the middle of each split, as *b b* in the figure, passes through the centre. Then in taking out a half brass, the lower edge marked *a* will describe an arc of a circle, having the shaft centre for a centre. Draw this arc and it will show the least distance from the centre to the inside of the pedestal, and will, therefore, give the thickness of the packing pieces. Notice that these

are made of the H form to save machining, and only bear against the squared part of the brasses and across the centre. Let flanges overlap pedestal about $\frac{3}{8}''$. Then draw in cap thickness and studs (see § 115), and finish cap. Show set screws for packing pieces and finish base, allowing a large radius for the fillet in consequence of the height of the shaft centre from the base. Notice other points as in § 116.)

(118) **Footstep Bearing.**—Vertical shafts are supported by resting their lower ends in footstep bearings. The lower part of the shaft is turned smaller to form a journal, and the end is turned slightly spherical, and rests upon a brass arranged in a suitable pedestal. As the pressure comes entirely upon the end of the shaft, it is somewhat difficult to ensure proper lubrication. In Fig. 159 is shown a footstep bearing for a $5\frac{1}{4}''$ diameter shaft, which is arranged to have continuous lubrication, and to allow of the brass being removed without having to lift the shaft. Experiments have shown that, by cutting a small groove (shown by dotted lines across a diameter of the shaft end) and introducing the oil under the centre of the shaft, then the shaft in rotating carries round a supply of oil in the groove, forces it radially outwards and up around the shaft until it issues at the top of the journal, and that if the supply under the shaft centre can be kept constant, then the lubrication will be very efficient, and the brass will carry a high pressure. The brass in Fig. 159 is arranged to effect this; it is of square form outside and recessed to form an oil cup, C C, the outside rim of the cup being carried up beyond the end of the journal in order to catch all the oil squeezed out. The oil then flows from the cup C C

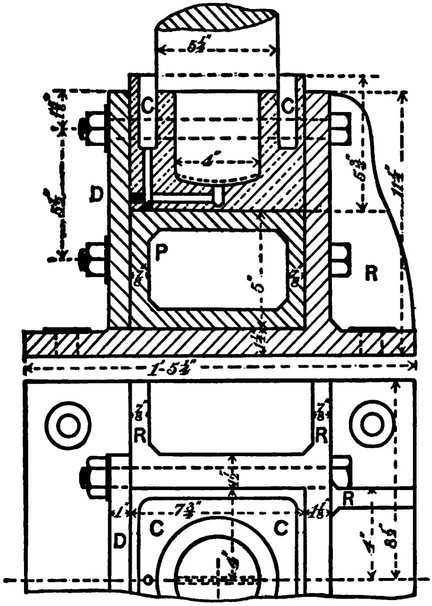


Fig. 159.

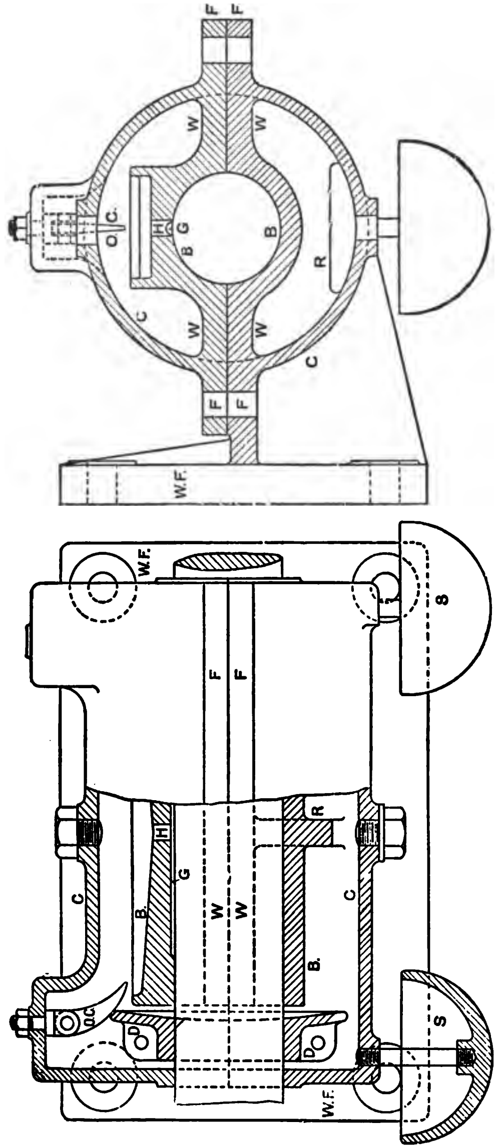


Fig. 160.

through the holes, as shown, to the centre of the shaft end. The oil holes require to be drilled in the brass, and afterwards plugged up, as shown. The brass rests upon a cast-iron box frame, P, having top, bottom, and two sides, which serves as a packing piece, and which when removed allows the brass to be slipped down from the journal and taken out, its depth must, therefore, slightly exceed the depth of the brass. The brass is surrounded by three sides of the cast-iron frame, having an enlarged base and stiffening ribs, R R, and secured by the cover D, fastened to the frame by the four bolts, as shown.

EXAMPLE

EX. 4.—Draw the footstep bearing, as in Fig 159, to the sizes given. Bolts in cover, $\frac{3}{4}''$ diameter; 1″ holding down bolts. Show bosses for holding down bolts. Scale 6″ = 1′.

(119) **Bath Lubricated Bearing.**—An example of a continuously lubricated shaft bearing for bolting to a pillar or wall is shown in Fig. 160. The length of the journal is three times the diameter, and in consideration of this and the full lubrication the bearing is of cast iron. The actual bearing B B which surrounds the shaft is enclosed in an outer casing O C, the lower half of which is attached to the flange W F for bolting to the pillar or wall. Each half of the bearing B B forms a part of its casing, being connected by the horizonal webs W W, and in the bottom half by the central rib R. The two halves of the casing are bolted together by the flanges F F at the back and front. At each end of the bearing B B cast-iron discs D, made in two halves, are bolted together round the shaft and turn with it, and above each disc a small curved clip, O C, is loosely jointed, as shown, so as to continually rest on the rim of the disc. Oil is poured into the casing by taking out the top set screw, and is carried round by the discs to be scraped off by the clips, down which it drips to the inclined top of the bearing and then through the hole H and groove G to the shaft. The bottom set screw is for removing old oil, and the cups or savealls S S are for catching any oil which oozes out between the shaft and the outer casing.

EXAMPLE.

EX. 5.—Make a working dimensioned drawing showing front elevation partly in section, end elevation in section, and plan of the bath lubricated bearing of Fig. 160, sizes as follows:—Shaft $3\frac{1}{4}''$ diameter. Bearing B B 10″ long, metal $\frac{5}{8}''$ thick, from centre to top of sides of inclined part in upper half $2\frac{7}{8}''$, outside width

$3\frac{1}{2}''$. Oil hole H $\frac{3}{8}''$ diameter, groove G $\frac{3}{8}''$ wide. Centre rib R $\frac{3}{4}''$ thick, hole at lowest part to allow oil to pass through. Casing $14\frac{1}{2}''$ long and $8\frac{1}{2}''$ diameter outside, thickness $\frac{3}{8}''$; webs W W $\frac{3}{4}''$, flanges $\frac{3}{4}''$. Flange for bolting to wall $1\frac{1}{8}''$ thick, $16\frac{3}{4}''$ long, $11''$ wide, $9''$ from centre of shaft. Projecting box corners for holding clips $3''$ outside in direction of shaft length, $2\frac{1}{2}''$ outside width, $5\frac{7}{8}''$ from centre of shaft to top. Cast-iron discs D $1\frac{7}{16}''$ total depth, thickness of metal next to shaft $\frac{7}{16}''$, length in direction of shaft $1\frac{1}{8}''$, other thickness $\frac{1}{4}''$, outside diameter of dished part $7\frac{1}{8}''$, fixed by $\frac{3}{8}''$ bolts, nearest face of disc $\frac{5}{16}''$ from end of bearing B B. Clips about $2\frac{1}{4}''$ long, $\frac{11}{16}''$ to $\frac{1}{8}''$ in width, $\frac{1}{4}''$ to $\frac{1}{16}''$ in thickness, fixed by $\frac{1}{4}''$ pins to a small forked rod, bolted through top of casing, as shown, rod $\frac{3}{8}''$ diameter. Oil cups $5''$ diameter, $2\frac{5}{8}''$ deep outside, fixed by $\frac{1}{2}''$ rods. Set screws for oil inlet and outlet $\frac{3}{4}''$ diameter. Scale $6'' = 1'$.

(Show projecting facing at ends of casing next shaft, and also bosses for set screws. Back flange to have centre strengthening ribs, as shown, and bosses for bolts. Flanges on casing need not extend for whole length, as two bolts on each side are sufficient.)

(120) **Wall Brackets.**—The bearings for workshop and mill shafting generally require fixing to walls or overhead beams, for which purpose a number of standard forms of cast-iron brackets are adopted.

The pedestal bearing may either be cast with the bracket, or bolted to it in the form of a separate plummer block, as in Fig. 162.

In Fig. 161 is shown the construction of a wall bracket for a $1\frac{3}{4}''$ diameter shaft, in which the bearing is cast with the bracket. As the pressure is all upon the bottom half of the bearing, there is no top brass, the shaft rubbing against the cast-iron cap. At first sight it would seem that the use of studs for holding the cap down would be more convenient than bolts, and would save the underneath bosses, but as previously mentioned, many engineers avoid studs owing to their tendency to break off where the screwed part ends. The view A is a section of the bracket at that part.

(121) **Savealls.**—In Figs. 160 and 161 cast-iron cups or boxes, S S, are shown on each side of the bearing. These are fitted to catch the waste oil oozing out at the ends of the brasses, and are generally made a part of all overhead shaft bearings. Many makers cast them on all pedestals, for whatever purposes, an example which the student will do well to imitate. They need to extend beyond the brass in both directions in order to catch all

the drops, and in thickness must not be less than $\frac{1}{4}''$, owing to difficulties of casting.

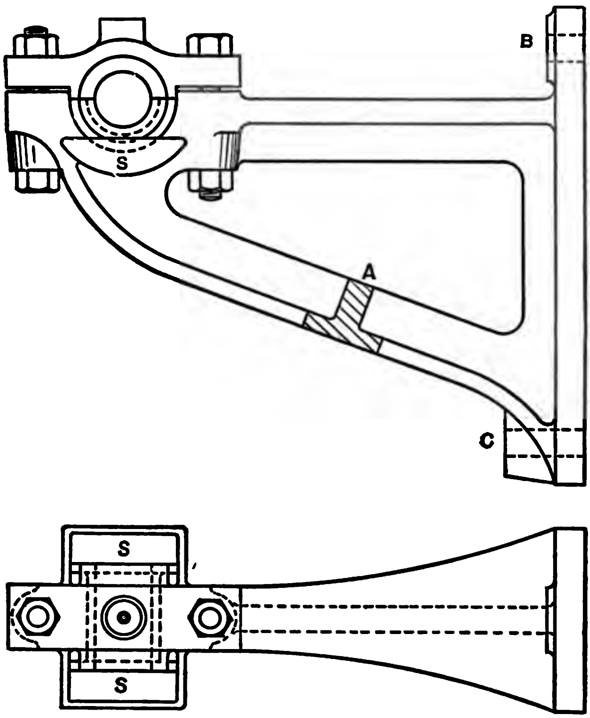


Fig. 161.

(122) **Overhead Bracket.**—A bracket for bolting to an overhead beam is shown in Fig. 162, the pedestal for a 2″ diameter shaft being a separate fitting bolted and wedged to the horizontal projecting arm. The square bosses B B are for the holding down bolts. The size of the top flange evidently depends upon the size of the beam, or the space available for supporting the bracket.

The calculation of the necessary amount of metal for such brackets as these is somewhat too difficult for a work of this character. But the student should notice that there are both bending and shearing forces acting upon the projecting arms, and that the T section of the arms, as shown at A, Fig. 161,

offers the most economical form for strength and stiffness, the latter being very necessary. On the fixing bolts of Fig. 161 there is a shearing stress acting on both bolts B and C, together with a tensile stress on the top bolt B, caused by the tendency of the bracket to turn downwards about the bottom edge of the flange

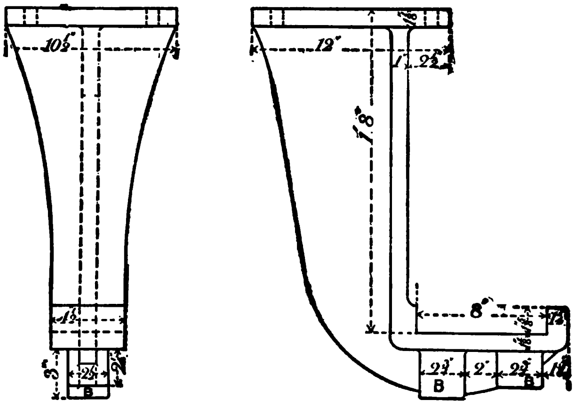


Fig. 162.

as a hinge. On the bolts of Fig. 162 there are tensile stresses only, hence other things being equal, the bolts for a bracket as in Fig. 161 should be larger than for a bracket as in Fig. 162.

It is doubtful if the makers of these brackets are able to calculate their necessary section, they are guided rather by previous experience, and the brackets are probably abnormally strong.

EXAMPLES.

Make drawings to a scale of 6″ = 1′, showing side and end elevations and plan of the following brackets for shaft bearings :—

EX. 6.—Wall bracket for a journal 1¾″ diameter, 3½″ long, as in Fig. 161, distance from centre to wall 14″. Pedestal part fitted with simple bottom brass with side flanges, width of cap and part through which bolts pass 2″, projection on both sides 2¾″ diameter to make up length of journal. Bolts for cap ⅝″ diameter, 5⅛″ centres, oil cup for glass needle lubricator, cast with cap. Savealls 4″ wide, 1¼″ deep, each 3″ from centre to outside in direction of shaft length. Wall flange 16″ long, 6″ wide, ⅞″ thick, inside web 1″ wide, ¾″ thick; top flange ¾″ thick, web 1¼″ wide; bottom flange ½″ thick, web 1½″ wide. Top and bottom flanges taper as shown from 6″ to 2″. Length of wall

flange above top flange 3″. Bosses for bolts in order to keep them close into top and bottom flange.

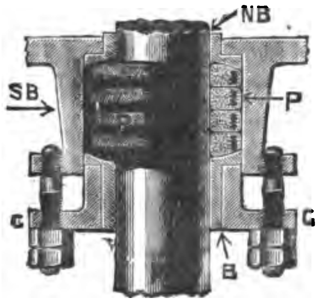
EX. 7.—Bracket for overhead beam as in Fig. 162 for 2″ shaft pedestal. Show pedestal in position, its design to be as in Fig. 154.

SECTION XXIV.

STUFFING BOXES.

THERE are a number of rods used in engines, and engine fittings of different kinds which require in working to move to and fro through the covers or casings of cylinders, or other vessels containing steam, gas, or air, without allowing any of the internal fluid to escape.

To prevent this leakage these rods require to pass through "*stuffing boxes*," which consist essentially of circular boxes containing rings of "*packing*" tightly surrounding the rods in such a way as to prevent the passage of fluid between it and the rod. The packing is kept in position and forced out against the rods by a loose bush called a "*gland*" held in place by studs and nuts (see Fig. 163*a*).



INDEX TO PARTS.

P R, Piston-rod.
S B, Stuffing box.
N B, Neck brass or back bush.
P, Packing (Bell's asbestos).
G, Gland.
B, Gland brass.

Fig. 163*a*.

For example, piston and slide valve rods of steam cylinders, pumps and rams for water pressure, spindles of cocks and valves, all require to pass through stuffing boxes, and as it is highly important to prevent leakage it is necessary for the student to understand the construction of stuffing boxes, and to be able to

design them in connection with any of the numerous parts to which they are fitted.

The number of patent stuffing boxes and different kinds of packing (metallic, hemp, asbestos fibre, &c.) at present in the market is far too large for description here. The student would do well to notice their chief features in the advertisements of the different engineering journals, but only general principles will be dealt with in the following remarks :—

In making drawings of stuffing boxes it is not usual to show the packing, or to state what kind it is to be.

(123) **Common Form of Stuffing Box.**—Fig. 163*b* shows a

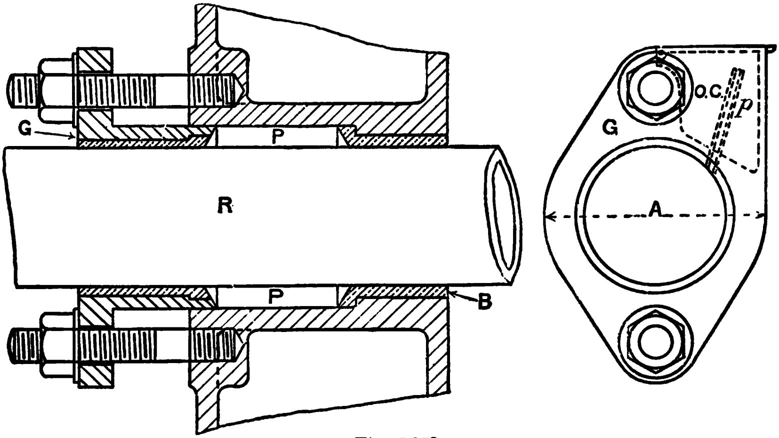


Fig. 163*b*.

stuffing box as fitted for the air pump plunger of a horizontal condenser (Fig. 197), which will serve as a good example of the most common form of construction.

The piece marked G is the "gland" which compresses the rings of packing in the space P P against the plunger R, when screwed in by the studs and nuts. For rods up to $2\frac{1}{2}''$ or $3''$ diameter this is generally made wholly of brass, but for larger sizes is of cast iron with a brass bush as shown in the figure. The most usual shape of the flange of the gland, when only two studs are used, is shown in the end view. Large boxes are fitted with three or four studs and circular flanges, and have frequently an arrangement of toothed gearing (a toothed ring on the gland piece and small pinions which form the nuts gearing into it), so that when one nut is turned all shall turn, and thus screw the gland in equally all round.

At the back of the box is fitted a brass bush, B, to keep the rod from rubbing on the hard metal of the cover or casing.

(124) **Lubrication of Stuffing Boxes.**—It is necessary to arrange for the rod and gland to be well lubricated. A convenient method when the box is horizontal is to extend one part of the gland flange and leave it hollow to form an oil cup, the oil being siphoned down the small supply pipe to the rod by a piece of cotton wick; a piece of thin brass hinged at one end forming a cover, this is shown in the figure, where O C is the oil cup, *p* the pipe, and L the lid.

With vertical boxes, lubrication is best effected by attaching a separate lubricator leading to a groove cut in the gland.

(125) **Proportions of Stuffing Boxes.**—These are generally decided by very practical considerations, the following being the most important:—

The length of a stuffing box for high pressures should be greater than for low pressures, whether steam or water, therefore the boxes of H.P. cylinders usually contain more packing than those of L.P. cylinders for the same diameter of rod.

Whenever possible it is always better to fix long stuffing boxes than short ones, they give far less trouble in working and require but little attention. Special limitations of space as in marine engines may sometimes handicap the designer, but generally speaking, a long box should be aimed at.

These same remarks apply also to the length of gland and of the back bush. They are subject to much wear, and, therefore, last longer when of good length. Frequently, as in the case of slide valve rods, they act as guides, and are made of extra length, especially when the rods do not extend through the back of the cylinder or casing, forming what are called "tail rods" (see Fig. 181).

The stuffing boxes of valves and cocks are of smaller size compared with the spindle diameter than those of cylinders and slide valve chests, as the spindle is only subject to occasional rotary movement, and the packing may be screwed up much harder against the rod.

The following proportions are taken from stuffing boxes on a number of engines and fittings, made by different makers:—

(126) **Size of Packing.**—Packing is made in sizes increasing by $\frac{1}{8}''$, but it is always sufficiently pliable to be compressed into slightly smaller spaces. The following sizes are measured between the rod and box, and, therefore, give all that is necessary to determine the diameter of the box, as, for example, a $1\frac{1}{2}''$ rod using $\frac{5}{8}''$ packing would require a box of diameter equal to $1\frac{1}{2} + (2 \times \frac{5}{8}) = 2\frac{3}{4}''$.

Diameter of Rod,	$\frac{5}{8}$,	$\frac{3}{4}$,	1,	$1\frac{1}{8}$,	$1\frac{1}{4}$,	$1\frac{3}{8}$,	$1\frac{1}{2}$,	$1\frac{3}{4}$,	$2\frac{1}{4}$,	$3\frac{3}{4}$,	$4\frac{3}{4}$ inches.
Size of Packing,	$\frac{1}{4}$,	$\frac{5}{16}$,	$\frac{3}{8}$,	$\frac{7}{16}$,	$\frac{1}{2}$,	$\frac{9}{16}$,	$\frac{5}{8}$,	$\frac{11}{16}$,	$\frac{3}{4}$,	$\frac{7}{8}$.	1 „

(A rule often used is, Size of packing $= \frac{1}{4}d + \frac{1}{8}''$ to $\frac{1}{4}''$ where $d =$ diameter of rod).

The above sizes are for steam pressures not exceeding 120 lbs. per square inch. For the stuffing boxes of air pumps and feed pumps, above 2″ diameter of rod, a smaller size may be used.

(127) **Depth of Packing.**—For piston-rod stuffing boxes, and for those of pumps having much movement, it is usual to put in enough rings of packing to give a depth of $1\frac{1}{2}d$ to $2d$—that is, 4 to 5 rings.

For the stuffing boxes of valves and cocks where the rods are only occasionally moved, the usual practice is to put in only two rounds of packing. The reason for this is that the gland can be screwed down much more tightly than with piston or pump rods, where the question of the friction of the stuffing box is important.

(128) **Length of Gland next Rod**

= $2d$ up to 1″ diameter.
$1\frac{1}{2}d$ from 1″ to 2″ diameter.
d to $1\frac{1}{4}d$ above 2″ „

(129) **Length of Bush next Rod**

= $\frac{1}{2}d$ up to 2″ diameter.
$\frac{3}{4}d$ above 2″ „

Thickness of brass liners for glands, or of bushes at back of box not less than $\frac{1}{4}''$.

(130) **Gland Studs and Flange.**—Where two studs are used, the diameter may be as small as $\frac{3}{8}''$ for the stuffing boxes of valves, but for piston or valve rods above $\frac{5}{8}''$ diameter the studs should not be less than $\frac{1}{2}''$. Studs of $\frac{5}{8}''$ diameter should be used for rods from 1″ to $1\frac{3}{8}''$ diameter, and of $\frac{3}{4}''$ diameter for rods from $1\frac{3}{8}''$ to $1\frac{3}{4}''$; above that size up to 4″ diameter, $\frac{7}{8}''$ or 1″ studs. Thickness of flange not less than diameter of studs. This gives a flange thickness of $1\frac{1}{4}$ to $1\frac{1}{2}$ times the thickness of the metal of the gland. The length of the studs must be sufficient to allow the gland to be fully out with the nuts on, and the length of the plain part of the studs should not be greater than the thickness of the gland flange. Lock nuts are frequently fitted. The ends of the gland and bush next the packing are generally turned at an angle of 60°, to act as wedges and force the packing out against the rod. The size across the gland in end view (marked A) need

only exceed the outside diameter of gland by about $\frac{1}{8}''$ to $\frac{1}{4}''$ on each side. When the box is external (Fig. 180) the size across the gland is not usually made less than the flange on the box.

All other sizes can be decided without difficulty after working previous examples.

(131) **Stuffing Boxes for Valves and Cocks.**—Fig. 164 shows the form of stuffing box commonly fitted to small valves and cocks. The outside of the box is screwed, to fit a covered hexagonal nut, N, which forces the small gland G down against the packing P. It is a good point to make the nut some standard size across the flats, a convenient size being that of a Whitworth nut one size less in diameter than the diameter of outside of stuffing box. The nut must be thick enough to allow of screwing on when newly packed, and is usually turned cylindrical above and below the hexagonal part as shown.

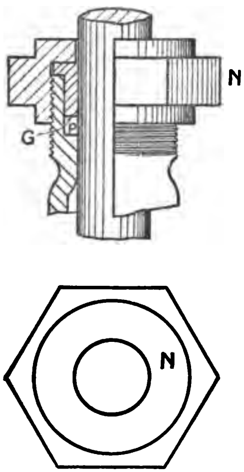


Fig. 164.

(132) **Stuffing Boxes for Hydraulic Purposes.**—For the rams and plungers of ordinary pumps, stuffing boxes with hemp packing of the same form as for steam engine work are used. Even when the water pressure is as high as 2,500 lbs. per square inch it is possible to keep the ram tight with ordinary packing, and in cases of rams having a slow but continuous movement it is customary to use only two or, at most, three rounds of packing, well compressed by the gland.

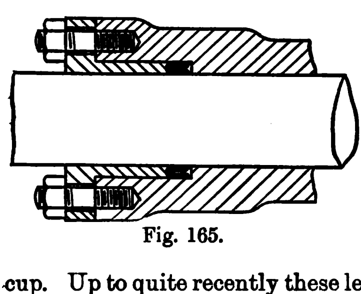


Fig. 165.

The special packing used for hydraulic rams of presses or testing machines was originally invented by Bramah, and consists of a cup leather of U section surrounding the ram and forced out against it by the water pressure inside the cup. Up to quite recently these leathers were made with a distance between the sides of the cup of about $2\frac{1}{2}$ times the thickness of the leather, but the latest make of ring is almost entirely closed, as seen in Fig. 165, with the sides of the cup practically touching. These

rings are fast displacing the older pattern as, besides taking up less space, they possess the great advantage of not working out of shape, and there really appears to be no reason why there should be any space between the sides of the cup. The box of Fig. 165 is for a hydraulic ram working under a pressure of 600 lbs. per square inch; the gland is screwed quite home, thus preventing the leather from having any movement.

(133) **Order of Drawing.**—In drawing stuffing boxes it is sufficient to show two views, one being in section, as in Fig. 163*b*. The gland should be shown out to its fullest extent. The order of drawing should be—first to decide diameter and length of box, length of gland and bush, diameter of studs, and then draw as follows:—

1. Centre lines, and lines showing rod in both views.
2. Line showing front of box, then inside and length.
3. Show back bush and gland piece.
4. Draw in studs and nuts (see § 42).
5. Finish gland in end view, the limits of flange being decided as previously described, and show arrangements for lubrication.

EXAMPLES.

Make working dimensioned drawings of the following stuffing boxes:—

EX. 1.—Stuffing box for air pump plunger of horizontal jet condenser (Figs. 163*b* and 197). Plunger, $3\frac{3}{4}''$ diameter; distance from outside of cover C to inside of division plate B, $7''$; thickness of metal, $\frac{1}{2}''$; metal around stuffing box, $\frac{5}{8}''$ thick; back bush, $2\frac{3}{4}''$ long; packing, $\frac{5}{8}''$. Scale, $6'' = 1'$.

(As this is for an air pump, the size of packing is less than for piston-rods; the back bush serves as a guide for the plunger. Bosses should be shown for the studs to prevent them entering the condenser; the projecting face on condenser is same shape as gland flange. Arrange for lubrication as shown.)

EX. 2.—Stuffing box for H.P. cylinder cover of vertical engine (Fig. 180). Rod, $1\frac{1}{4}''$; cover, $\frac{3}{4}''$ thick; metal of box, $\frac{5}{8}''$; flange on box for gland studs to be thick enough to give sufficient length of screw. Full size.

EX. 3.—Stuffing box for steam stop valve (Fig. 164). Spindle, $\frac{3}{4}''$ diameter; outside of box, $1\frac{5}{16}''$ diameter; screwed $1''$ gas thread, $\frac{3}{16}''$ packing; gland piece, $\frac{9}{16}''$, including flange $\frac{3}{16}''$ thick; depth of box, $\frac{5}{8}''$; nut, $\frac{7}{8}''$ total thickness; $\frac{1}{2}''$ of hexagonal part $1\frac{1}{16}$ across flats; cover, $\frac{1}{4}''$ thick. Full size.

EX. 4.—Stuffing box for hydraulic ram $3''$ diameter (Fig. 165).

Diameter of box, $3\frac{3}{4}''$; depth of box, $3''$; gland flange, $6\frac{1}{2}''$ diameter, $\frac{7}{8}''$ thick, $4-\frac{5}{8}''$ diameter studs; leather, $\frac{3}{4}''$ deep; casing of ram, $5\frac{1}{4}''$ diameter, increasing to $6''$ diameter for a length of $3''$, and to $6\frac{1}{2}''$ diameter for a length of $1\frac{1}{4}''$. Full size.

SECTION XXV.

COCKS AND VALVES.

Wherever pipes are used to convey steam, gas, or water, and in all machines worked by these fluids or employed in pumping, fittings are necessary to regulate the "*supply*" and "*discharge*." Such fittings consist of some form of "*cock*" or "*valve*" made in gunmetal or cast iron, the special construction and action of which depends upon the requirements of each case.

(134) **Cocks.**—In its simplest form, a cock consists of a hollow cylindrical plug, and a casing or shell in which the plug can turn round. Each is provided with two openings, and the plug

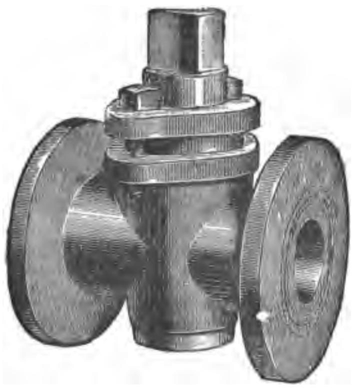


Fig. 166*a*.

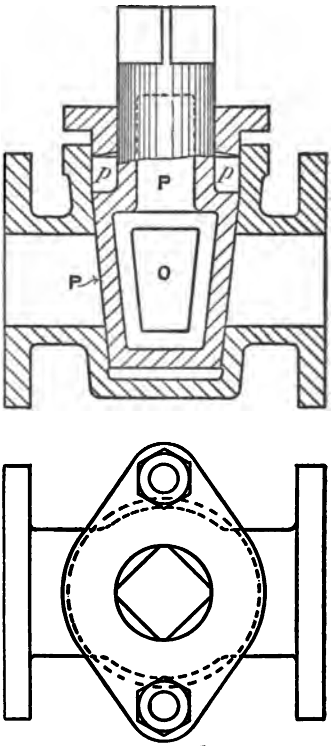


Fig. 166*b*.

can be so placed as to bring these openings opposite each other,

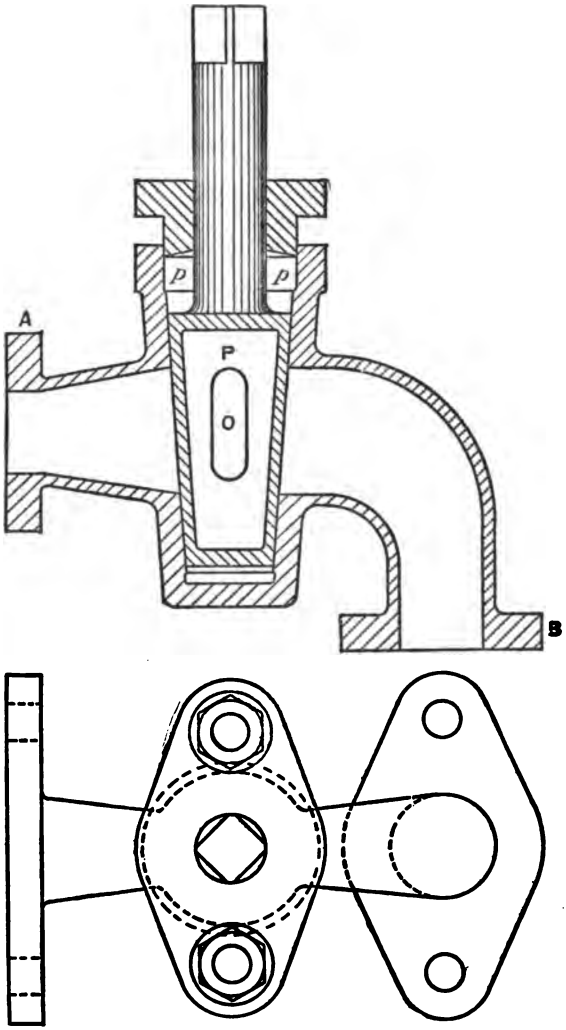


Fig. 167.

in which case, there is a clear passage way through the cock; or the openings in the plug can be brought opposite the solid part of the casing, thus shutting off the cock. The plug is slightly

conical, to ensure good fitting, and is provided with a stuffing box to prevent leakage. Flanges or sockets are cast on the casing for fixing to pipes or other parts.

The cock shown in Fig. 166*a* is of a pattern used for high pressure, steam or water, its detailed construction being shown by Fig. 166*b*. P is the plug shown in the position of "shut off," the opening being marked O, the spindle is in one piece with the plug, and is made with a square end to which a handle is fitted for turning the cock. The stuffing box for packing is marked *p*.

Fig. 167 shows a cock for water at ordinary pressure, having the inlet and outlet passages A and B at right angles.

The inlets and outlets of cocks may be made at any angle to each other to suit special needs, and small cocks are fitted with screwed ends or sockets instead of flanges. Sufficient points in the lines of interpenetration of the branches with the body of the cock, should be found as in Problem lvi., Fig. 90, to allow of the curves being properly drawn.

(135) **Three-way Cocks.**—These are provided with two outlets, having thus three ways. A sectional plane is shown in Fig. 168, the plug being placed to allow free passage from A to both B and C, and it will be easily seen that the plug can be turned to completely shut either B or C or both.

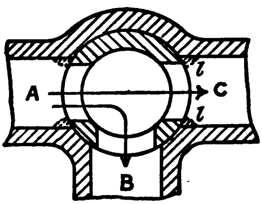


Fig. 168.

(136) **General Proportions and Construction.**—It will be recognised as a first principle that the area of the openings in the plug should each be equal to the area of the pipe, or of the waterway at the flanges. This obviously is not the case in the cock shown in Fig. 166, from which it follows that either a cock must be used of larger inside diameter at the flanges than the pipe to which it is fitted, or the water way of the cock and pipe must be unequal. It is to avoid this difficulty that the construction of Fig. 167 is sometimes adopted, the inlet and outlet are increased in the direction of the plug length as they approach the plug, and diminished in the opposite direction, thus attaining a rectangular section. This is desirable, for the further reason that when the openings in the plug are too wide, there is not metal enough left in the shell of the plug, except it be of large diameter, to overlap the openings in the casing sufficiently to prevent leakage, or the construction of casting inside lips as shown dotted at *l l*, Fig. 168, may be adopted. Evidently then, the first point in designing a cock for a certain size pipe is to

settle the proportions of the plug, and from that arrange the inlet and outlet branches, keeping the water way through each as nearly as possible equal to the pipe area. The following general proportions will assist in this :—

Diameter of waterway at flanges = diameter of pipe = d.
Diameter of plug at centre = d to $1\frac{1}{8}$ d.

Unless the construction of Fig. 167 or of the lips in Fig. 168 is adopted, this diameter must be arranged so that metal of plug sufficiently overlaps the shell around the opening to flanges in position of shut off.

Width of openings in plug = 0·6 d, height = 1·3 d for pattern of Fig. 167, ∴ area = $\cdot 78d^2$ = area of pipe.
Taper of plug on each side = 1 in 10.
Thickness of shell of plug = $\frac{1}{8}''$ to $\frac{7}{16}''$ up to 3″ diameter when of gunmetal, thicker when of cast iron.

Thickness is increased at top and bottom.

Length of plug should be arranged so as to overlap the casing at top and bottom, a distance of *not less* than $\frac{1}{8}$ d for light patterns, as in Fig. 167, and $\frac{1}{4}$ d for heavy patterns, as in Fig. 166.

Thickness of casing may be from $\frac{1}{8}''$ to $\frac{7}{16}''$ in gunmetal, and from $\frac{3}{8}''$ to $\frac{3}{4}''$ in cast iron, up to 4″ diameter. These proportions give abnormal strength, but are necessary for sound casting. Casing around plug is slightly thicker ($\frac{1}{16}''$ to $\frac{1}{8}''$) than at other parts, to allow for reboring. Diameter of plug spindle in gunmetal $\frac{3}{4}''$ to 2″, for cocks from 1″ to 4″; spindle made hollow for large sizes.

The proportions of stuffing boxes and flanges will present no difficulty after previous work. It is not necessary to put in more than two rounds of packing, hence the boxes need not be deep. Notice also the following points :—The plug should be drawn and fitted with the greater half of its length above the centre line, to allow of sinking when reground after wear, hence there must be clearance between the plug bottom and casing. The plug is generally turned by a handle fitting on end of spindle, which is made square for the purpose, this square being not quite perfect, as shown in the figures, to avoid sharp edges, and to make it easier to fit the handle. The flanges should be kept as near to the casing of the plug as possible, allowing clearance for a nut to be screwed on the bolts fastening the cock to the pipe. The holes in the cock flanges should be spaced so as not to bring a bolt directly at the top or bottom.

(137) **Order of Drawing.**—In drawing cocks it is usual to show two elevations and plan, the front elevation and half the plan being in section. Proceed in the following order :—

1. Draw centre lines of all views.
2. Draw plan of plug showing width of openings if as Fig. 166, and arrange diameter at centre so as to overlap when shut off, or use a smaller diameter plug and adopt the lips of Fig. 168.

3. Show plug in front elevation, tapering from centre line above and below; draw lines showing inlet and outlet passages, and decide length of plug, then finish plug.

4. Draw casing around plug and for stuffing box, also around the inlet and outlet passages.

(Increase diameter of spindle if gland comes too thick.)

5. Complete gland, showing studs and flanges.

(Gland does not require to be out from box more than $\frac{1}{4}''$ to $\frac{3}{8}''$ when fully packed, gland studs to be kept close in to casing of box, length of spindle 1″ to 2″ above top of studs, length of squared part equal to size across square.)

EXAMPLES.

Make working dimensioned drawings (three views) of the following plug cocks; full size.

EX. 1.—Cast-iron casing, gunmetal gland and plug, 3″ cock, heavy pattern, parallel passages for steam pressure, as in Fig. 166, shell of plug $\frac{3}{8}''$, spindle $1\frac{5}{8}''$, metal of casing $\frac{1}{2}''$, flanges $\frac{5}{8}''$ thick.

EX. 2.—Gunmetal cock $\frac{7}{8}''$, for water pressure, to pattern of Fig. 167; flange A, $2\frac{3}{8}''$ from centre of plug; centre of flange B, $2\frac{1}{4}''$ from centre; underside of flange, 2″ below centre; shell of plug, $\frac{1}{8}''$; thickness of metal around plug, $\frac{1}{4}''$ full; thickness at other parts, $\frac{1}{8}''$; spindle, $\frac{3}{4}''$ diameter; flanges, $\frac{3}{8}''$ thick. Inlet and outlet branches taper from $1\frac{1}{2}'' \times \frac{13}{16}''$ outside at casing of plug to $1\frac{1}{8}''$ diameter at flanges.

(138) **Valves.**—Of the large number of valves of different construction and manner of working examples of two classes only will be here dealt with: (i.) self-acting valves, such as used in pumps and operated by the action of the water; (ii.) valves opened and closed by hand, as in all stop and sluice valves for steam or water.

(139) **Indiarubber Disc Valve.**—Examples of this valve are shown in Figs. 169 and 170. An indiarubber disc, I.D., covers a gunmetal grating, G, the holes in which are arranged as shown in the left-hand half of the plan (Fig. 169), and is restricted in movement by a circular dished gunmetal guard, V.G., also provided with holes for lightness and to prevent the imprisonment of air or water between rubber and guard, shown above the rubber in section, and shown in plan in the right-hand half of the plan. Suppose this valve is fitted as the foot valve of a feed pump, then when the plunger moves upward the water passes through the grating from below, lifts the indiarubber, and enters the pump chamber; on the downward stroke of the plunger the indiarubber is forced down over the grating so that the water cannot return, and it is, therefore, pumped out through the

delivery valve. In this way the valve acts easily and efficiently, and it is largely used in condensers and in air pumps of large

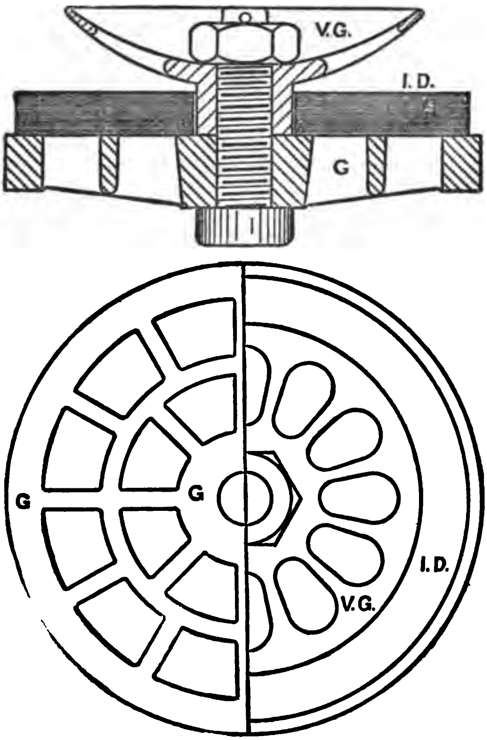


Fig. 169.

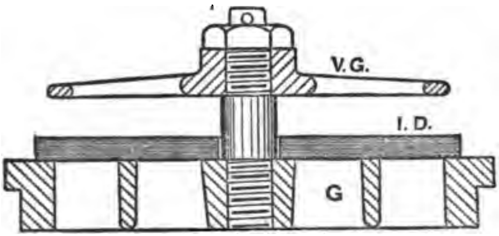


Fig. 170.

size (see § 208). It is generally necessary to design such a valve to have a certain area of waterway, and this is accom-

plished with sufficient exactness by allowing for the area of the division bars and the bolt boss, the effective area is about 0·4 to 0·6 of the area supposing the bars and centre boss not to exist. The division bars are generally from $\frac{3}{16}''$ to $\frac{1}{4}''$ thick, and are only necessary to support the rubber. Notice that the bars and boss in the grating should slightly taper to allow of drawing the pattern before casting, also that the thickness of the nut may be less than the ordinary standard. The diameter of the guard is about 0·7 to 0·8 the diameter of the rubber. Another construction of the valve is shown in Fig. 170, where the guard is flat, and fixed to the grating by a stud, the plain part being larger in diameter than the screwed part.

(140) **Lift Valves.**—Ordinary lift valves (Figs. 171*a* and *b*) are often used in pump valve boxes in place of rubber disc valves.

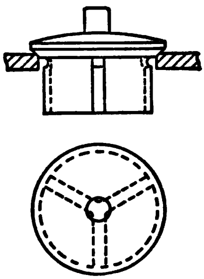


Fig. 171*a*.

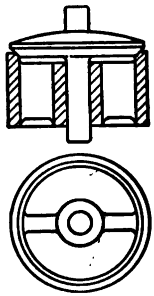


Fig. 171*b*.

They consist of discs of metal fitting over circular holes, the meeting surface or "seat" of the valve being turned slightly conical at an angle of 45°. These valves rise and fall with the strokes of the plunger exactly as the rubber disc valve of the last example, their movement being regulated by projections, which act as stops against the bosses on the top of the valves. It is necessary to fix guides to the valves to keep them over the seating, the two most common forms being shown in the figures. In Fig. 171*a* the valve is fitted with three "*guide blades*" or "*wings*," thin ribs of metal turned on their edges to a diameter slightly less than the inside diameter of the valve seat. These wings are sometimes curved in such a way as that the water impinging upon them gives the valve a slight rotary movement, and so brings it constantly in contact with different parts of the seating. In Fig. 171*b* the seating is a separate piece from the valve box, and has cast with it a hollow boss held in position by a thin central web, the valve being provided with a round rod

guided in the boss of the seating. A simple and efficient lift valve is obtained by using a hollow metal ball.

(141) **Lift of Disc Valves.**—By the diameter of the valve is understood the diameter of the hole at the seating, let this equal d, and let h be the height to which the valve should lift, in order to have an equal waterway round the valve and through the valve; then neglecting the valve guides:—

Waterway round valve = Waterway through valve.

$$d \times \pi \times h = d^2 \times \frac{\pi}{4}$$

$$\therefore h = \tfrac{1}{4} d.$$

Hence the lift of a valve must not be less than one-quarter of its diameter even when without guides, with guides its minimum lift may be of course less than this. As a general rule, the lift of stop valves is in excess of this. In metallic lift valves, such as in Fig. 171 where the movement is rapid, it is very necessary to have a small lift, otherwise the valve and seating will soon be damaged beyond further use by the force of the blows. In such cases a large valve with small lift is fitted. Such rapidly lifting valves may also have a wider seating than in stop and sluice valves lifted by hand, where the seating at first fitting does not exceed a width of $\frac{1}{8}''$.

(142) **Valves Lifted by Hand.**—These are made wholly

Fig. 172a.

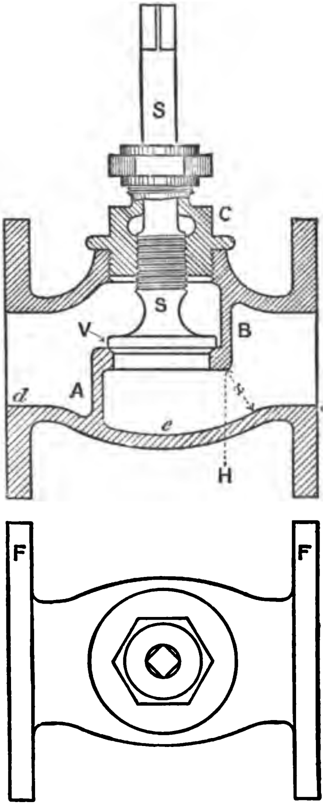


Fig. 172b.

of gunmetal, or the casing and cover is of cast iron. An ordinary type of gunmetal steam stop valve is shown in Figs. 172*a* and *b*; the valve V is in one piece with the spindle S, is therefore without guides, and is moved by the screwed part of the rod working in the cover C as a nut, the rod being kept steam tight by a small stuffing box, S B, a section of which is shown in Fig. 164 (sizes in Ex. 3, Section xxiv.) The division plates, A and B, and the plate forming the seating extend across the casing, and, therefore, prevent any passage of fluid except when the valve is open. Larger sized valves, and valves made of cast iron, have a flanged cover instead of a screwed cover as in the example. The flanges F F are for bolting in positions to pipes, &c.

It is not generally advisable to have the valve and valve spindle in one piece, as in the case of the spindle and the seating not being at right angles, the valve will not bear on all parts of the seating. This is entirely prevented by having the valve loose on the spindle

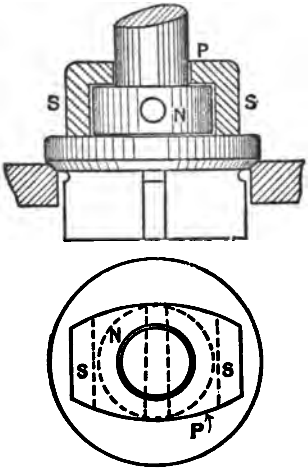


Fig. 173.

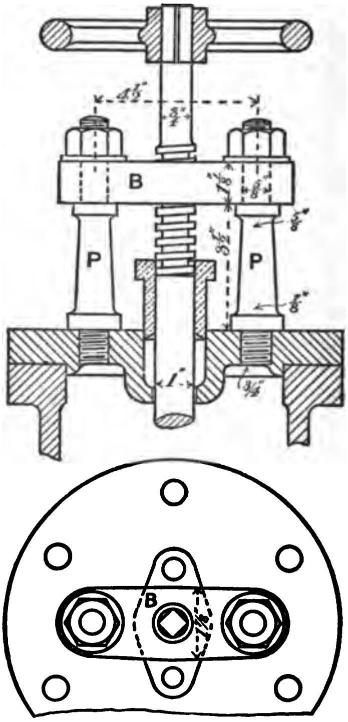


Fig. 174.

and free to set itself to the seating, although it is then necessary to fit the valve with guides. A common arrange-

ment is shown in Fig. 173. On the top of the valve is cast two upright strips, S S, connected by a top plate P, through which a hole is drilled slightly larger than the spindle; the end of the spindle is screwed, a round nut or collar, N, is slipped into position as shown, and the spindle is screwed into the nut from above, a taper pin being then driven through collar and spindle to make all firm. There should be clearance all round the nut, so that the valve has a slight movement independently of the spindle.

In valves of above 2″ diameter, especially when made of cast iron and used for water, the screw is placed outside the casing, and is, therefore, not exposed to rust. A usual design is shown in Fig. 174, the cover of the valve being provided with two turned uprights or pillars, P P, holding in position a crossbar, B, which serves as a nut for the screwed part of the spindle. The sizes given in the figure are taken from a 4″ diameter cast-iron valve, but the proportions for other sizes need present no difficulty. The crossbar should be designed in width as a nut and in depth equal to $1\frac{1}{8}$ to $1\frac{1}{4}$ times the diameter of the screw, the length of the uprights and their distance apart being arranged to give convenient clearance for the stuffing box.

If it is desired to prevent the holes in the cover for the pillars entering the valve chamber, bosses must be cast on the under side of the cover beneath where the holes will be drilled, or they may be cast above the cover, as outside bosses, thus shortening the necessary length of the pillars. In this type of valve for rough work, the crossbar and pillar is replaced by a cast-iron bracket bolted to the cover, and having a gunmetal bush to form a nut for the screw.

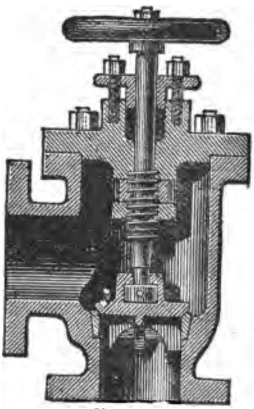


Fig. 175.

(143) **Valve with Passages at Right Angles.**—Another type of valve for connection to pipes at right angles, as is common for boiler stop valves, is shown in Fig. 175. When the casing is of cast iron the valve and seating are made of gunmetal, the seating being a separate fitting usually of the pattern of Fig. 171*b*, the under side of the cover for an inside screw has a gunmetal piece screwed into it to form a nut for the thread on the valve spindle. The central chamber should be of such an inside dia-

meter as that the waterway around the valve when off the seating, is equal to the waterway at the seating—that is, if d = diameter of valve and D = diameter inside valve chamber, then $\frac{\pi}{4}(D^2 - d^2) = \frac{\pi}{4} d^2$, or $D = \sqrt{2 d^2} = 1{\cdot}414\, d$.

The following general remarks will assist in designing valves of the construction described:—

The clear waterway through the valve should have an area equal to the area of the pipe, therefore when the valve has guides its diameter should be greater than the pipe diameter.

The diameter of inlet and outlet branches equals diameter of pipe.

The division plates, A B, Fig. 172*b*, are better sloping at about 45° than parallel to the valve spindle, and should be arranged to give equal space above and below the seating, therefore the centre line of the seating should coincide with the centre line through the branches.

The inside diameter of the cover for the pattern of Fig. 172 need only be sufficient to admit the valve freely.

The distance above the seating to the under side of the cover must allow of the under side of the valve being *at least* one quarter its diameter above the seating when fully open. It is generally in excess of this.

The diameter of the casing around the seating for the pattern of Fig. 172 must be such that the distance x in Fig. 172*b*, multiplied by the mean size across the casing inside in a direction at right angles, is equal to the area of the pipe. This means that inside diameter at the section through the line B H in the figure must equal $1{\cdot}41\, d + \frac{1}{2}$ thickness of seating.

For the pattern of Fig. 172*b* it is of advantage to consider the casing as symmetrical about the centre lines before the cover part is fixed, as this makes it easier to construct the pattern, therefore the top lines should first be drawn coinciding with the bottom lines $d\,e\,f$, and then the circular part for the cover added. The drawing is slightly inaccurate in this respect, but the idea is clearly shown in Fig. 172*a*.

The flanges should be as close in as convenient to allow for the nuts of the fixing bolts not meeting the casing, and should be proportioned exactly as in flanged joints.

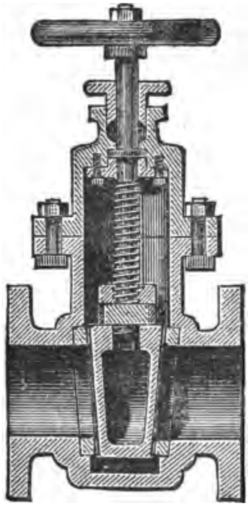
Valve spindles should be of wrought iron or mild steel, screwed with a square thread.

The valve is often arranged to have the pressure above it when closed, as this assists in keeping the valve tight; on the other hand, a valve is easier to open when the pressure is below it, and the gland is not so likely to leak.

The design of other parts, such as the stuffing boxes, flanges, spindle, and size of handwheel, should present no difficulty.

(144) **Sluice Valves.**—These are chiefly used for water, and in construction agree with the example of Fig. 176. The valve is a hollow cylindrical box, slightly tapered as the plug of a cock moving between guides, and for a full waterway it requires to be lifted a distance equal to its diameter. For this reason this type is not largely used for steam work, although in other respects their construction possesses good features. Notice that the

valve moves up and down the spindle, which simply turns round, hence the necessity for the collar shown under the stuffing box. These valves are also made with outside screws.



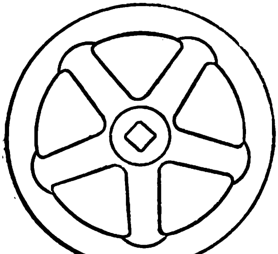


Fig. 177*a*.

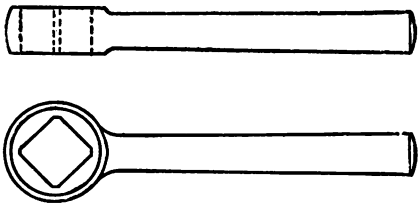


Fig. 176. Fig. 177*b*.

(145) **Handwheels and Handles** (Figs. 177*a* and *b*).—These are generally tightly driven on the square end of the spindle, but may be held on by a nut if necessary. The proportions of cast-iron handwheels and wrought-iron handles are given in the following table. A wheel of $7\frac{1}{2}''$ diameter, having five arms, is shown in section in Fig. 174, and in plan in Fig. 177*a*. Cocks are fitted with handles, and valves with handwheels :—

Sizes of Handwheels (Figs. 174 and 177*a*).							Handles (Fig. 177*b*).		
Diameter Outside Wheel.	Boss.		Diameter of Rim.	Arms.		No. of Arms.	Diameter of Cock.	Handles.	
	Diameter.	Thickness.		Width.	Thickness.			Length.	Mean Diameter.
4″	1″	$\frac{5}{8}''$	$\frac{9}{16}''$	$\frac{7}{16}''$	$\frac{5}{16}''$	4	1″	5″	$\frac{1}{2}''$
$5\frac{1}{4}''$	$1\frac{1}{4}''$	$\frac{11}{16}''$	$\frac{5}{8}''$	$\frac{1}{2}''$	$\frac{3}{8}''$	4	$1\frac{1}{2}''$	6″	$\frac{5}{8}''$
7″	$1\frac{3}{4}''$	$\frac{7}{8}''$	$\frac{13}{16}''$	$\frac{3}{4}''$	$\frac{7}{16}''$	4	$2\frac{1}{2}''$	$7\frac{1}{2}''$	$\frac{13}{16}''$
9″	2″	1″	$\frac{15}{16}''$	$\frac{13}{16}''$	$\frac{1}{2}''$	4	3″	10″	1″
11″	$2\frac{1}{4}''$	$1\frac{1}{8}''$	$1\frac{1}{16}''$	$\frac{7}{8}''$	$\frac{9}{16}''$	4	4″	11″	$1\frac{1}{16}''$

(146) **Drawings of Valves.**—In drawing valves of the indiarubber disc type (Figs. 169 and 170) it is sufficient to show two views, a sectional elevation and a plan, one-half of the plan to show the grating with the indiarubber removed, and the other half the guard. With stop and sluice valves (Figs. 172, 175, and 176) it is usual to draw two elevations and a plan, the front elevation and sometimes half the plan being in section. The order of drawing a valve of the type of Fig. 172 should be as follows, the proportions being first decided:—

1. Draw centre lines of all views.
2. Draw seating in front elevation, to be bisected both by the centre line of the valve, spindle, and the centre line of the inlet and outlet branches.
3. Draw valve and spindle in position when closed.
4. Draw in inside and outside of valve casing on both sides of the centre line (this is to ensure a symmetrical section).
5. Draw in cover and stuffing box, allowing for clearance between cover and valve, and for a clearance above the screwed part of the spindle equal in length to the valve lift.
6. Draw division plates, flanges, and complete details.

EXAMPLES.

Make working dimensioned drawings of the following valves:—

EX. 3.—Indiarubber disc valve with dished guard (Fig. 169). Diameter of rubber, $8\frac{3}{4}''$; thickness, $\frac{3}{4}''$; depth of grating, $1\frac{3}{8}''$; diameter inside, $8''$; outside, $9\frac{1}{2}''$; division bars, $\frac{3}{8}''$ thick at top; guard, $6\frac{1}{2}''$ diameter outside, $1\frac{1}{2}''$ extreme height; thickness, $\frac{3}{16}''$ to $\frac{5}{16}''$; screw for guard, $1''$ diameter. Calculate effective area through grating. Full size.

(Arrange grating with eight division bars in inner circle, and twelve in outer circle. Boss of grating and guard to suit screw, and to give overlap for rubber on boss of grating. About $\frac{1}{8}''$ to $\frac{3}{16}''$ clearance between top of rubber and guard. Ten or twelve holes in guard.)

EX. 4.—Gunmetal steam stop valve for pipes of $2''$ diameter to pattern of Fig. 172, screwed cover; valve, $\frac{3}{8}''$ thick; spindle, $\frac{3}{4}''$ diameter, valve and spindle in one; general thickness of metal of casing, $\frac{5}{16}''$; distance outside flanges, $6\frac{1}{2}''$. Full size.

(Spindle enlarged in diameter at screwed part; length of screw in cover forming nut for spindle not less than $\frac{3}{4}''$ diameter; nuts on cover and gland to be some standard size; proportions of stuffing box given in Ex. 3, Section xxiv.)

EX. 5.—Cast-iron steam stop valve for pipes of $4''$ diameter with outside screws and flanges at right angles (Fig. 175).

Gunmetal valve, seating, and gland, wrought-iron spindle, 1″ diameter. Scale 9″ = 1′.

(Seating as in Fig. 171*b*, valve as Fig. 173, proportions of arrangements for outside screw in Fig. 174, top of valve seating on level with horizontal outlet, flanges as close in as convenient for nuts and proportioned same as for cast-iron pipes (see §§ 44-46); casing thickened around seating—see Problem lv. for method of obtaining points in the intersection of the horizontal branch with the main casing.)

STEAM ENGINE DESIGN.

SECTION XXVI.

INTRODUCTION.

THE object of the following sections is to give an elementary knowledge of the principles and practice of steam engine design as far as the construction of simple single-cylinder engines of ordinary type and of small power. It will be assumed that something is known of the uses of the different parts of such engines, and how the work of an engine is measured. The limits of the book will not permit of anything like a full consideration of the conditions which decide the proportions of many of the parts, such as connecting-rods and cranks, owing to their requiring a somewhat advanced treatment; but as far as possible the reasons for the sizes and arrangement adopted will be made clear, both from the standpoint of the stresses the parts have to stand and the practical requirements of construction. No consideration will be given to the design of such simple parts as have been dealt with in the preceding sections, it being presumed that such sections have been worked through, or will be referred to.

The majority of the illustrations are those of the parts of two separate engines, each constructed by eminent engineering firms, and each typical of their class. But in some cases additional designs are illustrated taken from engines constructed by other makers, so that a comparison can be made of the relative advantages and disadvantages of different arrangements of the same parts.

The questions appended to each section are so arranged that when all the "Examples A," or all the "Examples B," have been worked, the student will possess drawings of the leading parts of the following engines :—

(147) **Examples A.**—A high-speed vertical engine, having the cylinder inverted above the crank, and suitable for the direct driving of pumps or dynamos. The cylinder is supported by two turned steel columns in front, and by a cast-iron standard behind, which forms part of a framing of box section, containing the crank-shaft bearings. The crosshead and guide is of the single slipper pattern, working between guides bolted to the back standard. The slide valve is of the piston valve type, worked by an eccentric placed outside the shaft bearing. The crank is made with solid counter-balance weights. Diameter of cylinder $7\frac{1}{2}''$, stroke $5\frac{3}{4}''$, revolutions per minute 400, about 35 I.H.P., with a boiler pressure of 115 lbs. per square inch.* The general arrangement of the engine is shown in Fig. 178. (*See Folding-plate.*)

(148) **Examples B.**—A horizontal engine, having a jet condenser of cast-iron box pattern, the cylinder and condenser being bolted to the same bed-plate, and suitable for the ordinary driving of machinery. The cylinder is steam jacketed, and is fitted with a main slide valve and an expansion valve, the eccentrics for which are fixed inside the crank-shaft bearings. The crosshead is attached to double guide blocks, working between double guide bars. The piston-rod is extended beyond the back cylinder cover, and works the air pump, which is of the plunger pattern. Diameter of cylinder $8\frac{1}{2}''$, stroke 12″, revolutions per minute 150, about 25 I.H.P., with a boiler pressure of 105 lbs. per square inch. The general arrangement of the engine is shown in Fig. 179. (*See Folding-plate.*)

It must, therefore, be understood that all future references to Ex. A or Ex. B relate to these two engines. The questions forming the examples of each section are worded to show the relation of the different parts, as a result of which it will be necessary to refer to previous examples or to §§ 147, 148, in order to obtain all the necessary particulars. For instance, in working Ex. A 2, Section xxvii., the question of Ex. A 1, must be referred to in order to see the size of the cylinder to which the piston is to be fitted. It is only in this way that the interdependance of the several parts can be fully kept in mind.

(149) **Order of Design.**—The first proceeding in designing any complete machine is to decide, at least approximately, the chief sizes, form and arrangement of the leading parts. This is especially necessary in the case of a steam engine, as a desired power can be obtained from quite different proportions of the two leading dimensions, the piston diameter and the length of the stroke; and as the sizes of many parts depend so much upon the sizes of other parts; while equally good engines can be con-

* All steam pressures are stated as "pounds per square inch absolute."

ONTAL STEAM ENGINE

only.	C
Line *a-b*.	C.(
Line *c-d*.	S.(
	S.C.(
ank-shaft bearings.	S.]
nterbalance weights on cranks.	E.]
ve-rod gland.	P.]
ve-rod.	(
de bracket for valve-rod.	G.]
entric.	C.]
nt columns.	
k standard.	
ming.	
nge for cylinder on back standard.	

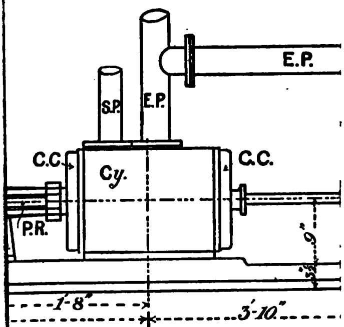


Fig. 179.

structed with quite different forms and arrangements of many of the parts.

It would be too lengthy a matter to describe fully the methods adopted in getting out the designs for a steam engine, and not altogether a useful one in our case, since we are unable to carry our investigations far enough to make our knowledge of the methods of much value; but a brief indication of the usual plan of procedure may not be out of place.

We know that the method of finding the horse-power of an engine is as follows:—

$$\text{H.P.} = \frac{\text{mean effective steam pressure on piston} \times \text{area of piston} \times \text{length of stroke} \times \text{number of strokes per minute}}{33{,}000}$$

or, if p = mean steam pressure on the piston in lbs. per sq. inch.
d = diameter of the piston in inches.
s = length of stroke in feet.
N = number of revolutions per minute,

then

$$\text{H.P.} = \frac{p \times d^2 \times \frac{\pi}{4} \times s \times 2N}{33{,}000}$$

Hence the designer must settle the steam pressure, the piston diameter, stroke, and speed; and we see that he can obtain the same power with many different values of the four factors. The boiler pressure at which the engine is to work is first settled, the usual pressures being 85 lbs. per square inch for non-condensing, and 60 lbs. per square inch for condensing engine, for ordinary small sized engines.

An indicator diagram is then drawn, satisfying the conditions of steam distribution such as will be arranged for in the engine, and allowing for the loss in pressure between the boiler and the cylinder. From this diagram the mean working steam pressure can be found, and thus one factor in the horse-power settled. The next step is to decide the relation between the piston diameter and its stroke, and the revolutions per minute, and then find the diameter or the stroke which, with the other factors, will ensure the required power, allowing for the loss of work in driving the engine itself if the power stated is effective horse-power required. The conditions which help to decide the piston diameter, stroke, and speed are as follows:—

(150) **Piston Speed.**—This is measured by the number of feet passed through by the piston per minute, and is therefore equal to the piston stroke multiplied by number of strokes per minute, or $s \times 2N$. A high piston speed increases the wear and tear of the moving parts, and produces large stresses in the

engine framing, hence a low-piston speed is more desirable than a high one. The usual piston speeds adopted in practice for the class of engines we are considering are 240 to 300 feet per minute for ordinary driving, at from 120 to 180 revolutions per minute, and 400 to 500 feet per minute for higher speed engines, running up to 400 revolutions per minute.

(151) **Relation of Piston Diameter to Stroke.**—Owing to the advantages of a low-piston speed, engines running at a great number of revolutions per minute are generally made with a shorter stroke compared with their diameter than engines running at the ordinary speed. If a high-speed engine has a long stroke, then as the length of the crank is equal to half the stroke, the speed of the crank pin will be considerable, and will thus make the design and easy working of the brasses in the crank pin end of the connecting-rod more difficult, while it will also entail a larger connecting-rod, and thus increase the length of the engine.

The length of stroke may be limited by such practical considerations as want of space, as, for example, in marine engines for warships, where, owing to the machinery having to be kept below the water line, the piston strokes are always small compared to their diameters.

An important point in deciding the question is, that increasing the piston diameter increases the total pressure upon the piston, and, therefore, upon all its attached parts, such as the piston-rod, guides, blocks, connecting-rod, and crank, and thus necessitates their stronger construction; whereas an increase in the stroke only affects the stresses in the crank shaft.

The relation between the piston diameter and stroke in engines of these types is as follows:—

Horizontal engines, revolutions per minute from 120 to 180, the stroke varies from one and a-half to twice the piston diameter, the usual standard being twice.

Vertical engines, for the same range of speed, the stroke varies from $1\frac{1}{4}$ to $1\frac{1}{2}$ times the piston diameter.

For higher speeds, up to 400 revolutions per minute, both with horizontal and vertical cylinders, the stroke is as low as from 1 to 0·75 times the piston diameter.

In the engine of Ex. A the ratio is $\frac{5\cdot75}{7\cdot5} = 0\cdot76$, and in Ex. B $\frac{12}{8\cdot5} = 1\cdot4$.

Having then decided the steam pressure, the number of revolutions per minute, and the ratio of the piston diameter to the stroke, the designer can substitute the values so found in the

formula already given for the horse-power, and thus find the piston diameter and stroke, which, with the other conditions, will ensure the required power. He is then in a position to make detail drawings of the separate parts, proportioning them, as we shall see, to withstand the stresses produced by the steam pressure upon the piston.

SECTION XXVII.

CYLINDERS—PISTONS—SLIDE VALVES AND RODS.

(152) **Cylinders.**—A steam engine cylinder is made of cast iron, and in its simplest form may be described as consisting of a cylindrical barrel, having on one side of it the four sides of a rectangular box, called the "*steam chest*" or "*valve chest*," from which rectangular passages called "*steam passages*" lead, one to each end of the cylinder. The cylinder when for a vertical engine is cast with one end attached, or may be fitted with separate covers at both ends, stuffing boxes being provided for the passage of the piston-rod. The inside of the barrel is accurately bored to give a working surface for the piston, and in some cases is fitted with a separate "*liner*," the advantage being that by simply providing a new liner an old cylinder is made practically equal to a new one, whereas a cylinder without a liner, and too thin for further reboring, would need to be completely replaced. When a liner is used it may be of harder metal than the cylinder casting, and is often made of steel, thereby ensuring a more durable surface than with ordinary cast iron. A liner helps to make a simpler casting when the cylinder requires to be steam jacketed—that is, to be provided with a double casing (except at the steam chest), through which a supply of steam may pass when the engine is working, as shown in the cylinder of Fig. 181. It will be seen from the same figure that the cylinder covers are also cast with a double thickness, and having a space between for steam jacketing, so that the ends may be kept as hot as the sides. Simple engines are not usually steam jacketed, but this cylinder is included because of its value in showing the construction adopted when a liner and steam jackets are fitted. The cylinder casting is usually provided with projecting flanges

forming the feet by means of which the cylinder is bolted to the engine framing; in vertical engines these are cast on one end, as in Fig. 180, and in a horizontal cylinder on the under side of the barrel, as in Fig. 181.

The steam chest is provided with a separate cover, and one or both ends are made with stuffing boxes for the passage of the slide valve-rod. That part of the steam chest from which the steam passages lead to the cylinder is called the "*port face*," the openings to the passages being "*steam ports*." Between the two steam ports, a third and larger port called the "*exhaust port*" is provided, through which the exhaust steam from the cylinder passes away. The exhaust port opens into the space between the barrel and port faces which thus forms a convenient outlet for the exhaust steam.

The form of the steam chest and the arrangement of the ports depend upon the type of slide valve used. For the simple and most common valve, the D valve, the valve chest is rectangular and extends the whole length of the barrel, the port face being a flat surface. Its size also depends upon whether one or two valves are used. The valve chest of Fig. 181 is made for an ordinary valve and an expansion valve, and is thus provided with stuffing boxes for two rods. When the type of valve known as the "*piston valve*" is used, as in Fig. 180, the valve chest is of almost circular section, and the steam ports are rectangular openings cut in the circular barrel, the passages for the initial and exhaust steam being somewhat different from that just explained, as will be described later.

From an inspection of Figs. 180 and 181 it will be evident that the sizes and shape of the cylinder depends upon the type of slide valve used, upon the distance of the valve centre from the centre of the cylinder, and upon the width and shape of the piston, so that we cannot completely draw the cylinder until we have decided these points. The reasons which decide the kind of valve and piston to be used will be discussed fully when treating of their design; for the present we will assume they have been decided, and then see how to design the cylinder. In practical design the sizes of these dependant parts would be decided before proceeding to the details of either.

It will be seen from the general arrangements of Fig. 178 that the distance of the slide-valve centre from the cylinder centre depends entirely upon the position of the eccentrics on the crank shaft. In this engine (Ex. A) the eccentric is obviously most conveniently placed outside the crank shaft bearing, and close up to it, hence its distance from the engine centre line cannot be fixed until the sizes of the crank pin, crank webs, and crank-

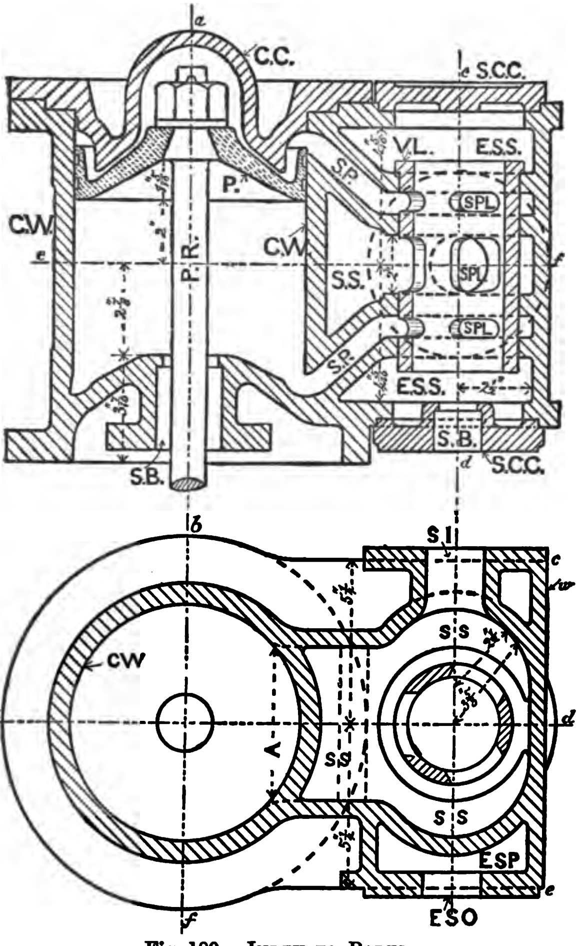


Fig. 180.—Index to Parts.

Sectional elevation and plan of vertical cylinder. (The plan is a section through the centre line *e f*.)

P. for	Piston.	S.C.C. for	Steam chest covers.
P.R. ,,	Piston-rod.	S.B. ,,	Stuffing boxes.
C.W. ,,	Cylinder walls.	S.I. ,,	Steam inlet.
C.C. ,,	Cylinder cover.	S.S. ,,	Steam space.
S.P. ,,	Steam passages.	E.S.S. ,,	Exhaust steam space.
S.P.L. ,,	Steam ports in liner.	E.S.P. ,,	Exhaust steam passage.
V.L. ,,	Valve liner.	E.S.O. ,,	Exhaust steam outlet.

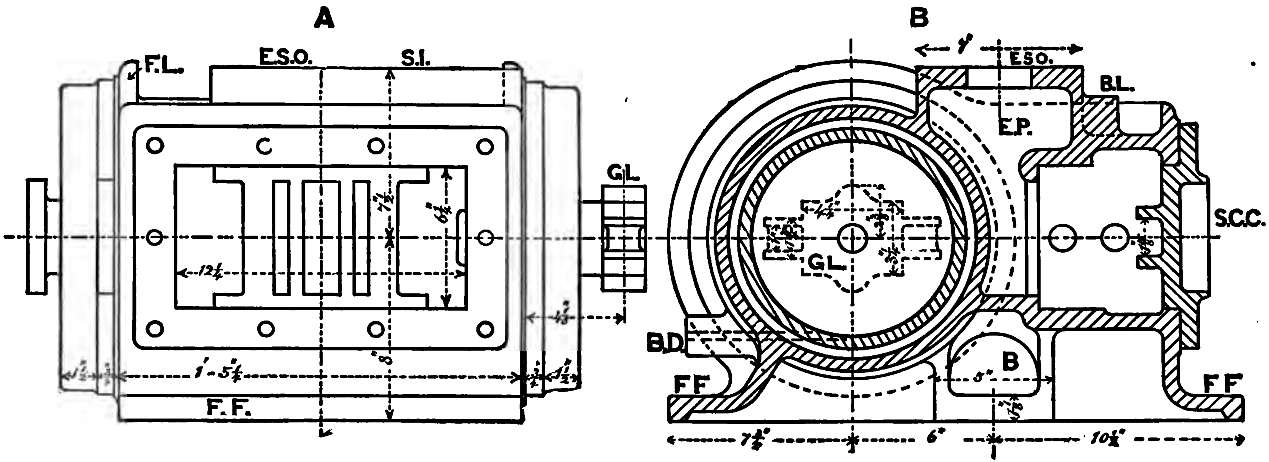


Fig. 181.

C

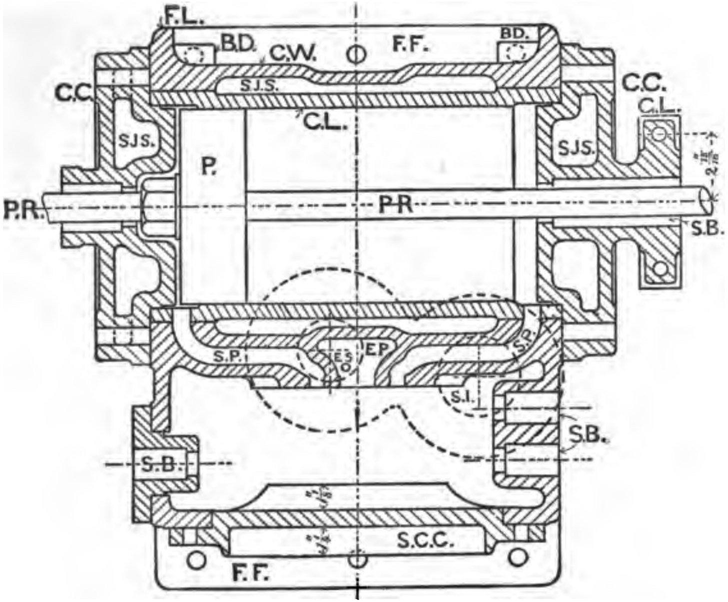


Fig. 181.

Index to Parts.

Sectional plan and end elevation and front elevation of vertical cylinder.

A = front elevation.
B = end elevation, a section through centre line.
C = plan in section.

P.	for	Piston.
P.R.	,,	Piston-rod.
C.L.	,,	Cylinder liner.
S.J.S.	,,	Steam jacket space.
C.W.	,,	Cylinder walls.
C.C.	,,	Cylinder covers.
S.P.	,,	Steam passages.
E.P.	,,	Exhaust passage.
S.C.C.	,,	Steam chest cover
S.B.	,,	Stuffing boxes.
S.I.	,,	Steam inlet.
E.S.O.	for	Exhaust steam outlet.
F.F.	,,	Feet for bolting to bed plate.
F.L.	,,	Flanges for lagging.
B.	,,	Bracket for supporting ends.
B.L.	,,	Boss for lubricator.
G.L.	,,	Lugs for guide bars.
B.D.	,,	Bosses for drain cocks.

shaft bearings have been decided. In the horizontal engine of Ex. B there is nothing to prevent the eccentrics being placed as near to the crank as possible, without fouling the guide bars, and, as will be seen by looking at the cylinder drawing, Fig. 181, this means that the port face may be as close in as is possible with other necessary proportions of the cylinder.

(153) **Vertical Cylinder with Piston Valve** (Ex. A, Fig. 180).—The steam chest extends for the whole length of the cylinder, and is provided with loose top and bottom covers, the bottom cover having a stuffing box for the valve-rod. The cross section inside, as seen in the plan, is rectangular with semicircular ends, the steam inlet being at the back, and the exhaust outlet at the front. The chest is divided horizontally by four plates forming a continuation of the walls of the steam passages, each plate having a central hole so that the liner may be tightly fitted in place, bearing against the edges of the holes. The liner has a small top flange, and contains openings at three different heights, which form the steam ports, the object of the liner being to allow of these ports being cut more conveniently and accurately than would be possible if the liner was cast with the cylinder. As seen in plan, there are three openings or ports at each height, the upper and lower ones communicating with the steam passages, and the centre ones with the space marked S S between the two middle division plates. The steam inlet leads into this space, so that the steam makes its way into the inside of the liner through the central openings, and then passes through the upper and lower openings into the top and bottom steam passages, as these are uncovered by the piston valve (see Fig. 185). The space marked S S thus serves as a steam reservoir.

The exhaust steam passes out from the cylinder through the same openings, but above and below the valve, and then escapes through the front passage marked E S P, to the exhaust pipe. This passage extends the whole length of the steam chest, and, therefore, the elevation of the outside of the chest is of oblong outline, and similarly for the side elevation, a web of metal being cast on the back of the chest (marked *w*) and connecting the top and bottom ends. The shape of these ends is shown in plan by the outline marked *a*, *b*, *c*, *d*, *e*, *f*, which may also be taken to represent the plan of the top of the cylinder casting with the covers removed, the plan of the bottom end being similar. The covers for the valve chest cover the circular openings in the top and bottom thickness of metal, and in outline are oblong with semicircular ends, as shown by the plan of the valve chest; the flange of the top cylinder cover is cut away next the valve chest,

to give room for the valve chest cover. It will be seen that the distance of the ports from the cylinder requires long steam passages, and, therefore, a considerable addition to the clearance, but this could only be avoided by using a large diameter of valve, which would necessitate a larger valve chest, or by arranging the valve and eccentric-rods, so that they are not in the same vertical line.

The lower flange of the cylinder is for bolting to the top of the engine standard.

(154) **Horizontal Cylinder, with Expansion Valve** (Ex. B, Fig. 181).—As already pointed out, this cylinder is fitted with a liner, and is provided with steam jackets, the steam from the cylinder jacket passing to the cover jackets through an opening not shown in the drawing. The piston-rod passes through both covers, in order to drive the air pump, the front stuffing box being prolonged and extended in order to provide lugs or projections to which the engine guide bars may be bolted. The port face is brought as near the cylinder centre as possible, allowing for the liner, jacket space, steam passages, and walls, and, as shown in the elevation and plan, has a projecting flat face, which can be readily machined to provide a good working surface for the valve. The expansion valve-rod passes through the back of the chest, and its stuffing box is a separate casting, having a flange bolted to the chest, which also forms a support for the expansion gear (Fig. 186). The top and bottom flanges of the chest are *outside*, and the two end flanges *inside*, thus giving a smaller cover, the inside ribs on the cover serving both to strengthen it and to prevent the expansion valve from moving from its seat.

A projection is cast above the valve chest, to which is bolted the steam and exhaust pipes. The steam enters the chest through the inlet, and makes its escape as exhaust through the exhaust port into the space marked E P, and thence to the exhaust pipe. The cylinder is bolted to the bed plate by the flanges F F, which extend for the whole length of the cylinder along each side. By looking at the general arrangement (Fig. 179) it will be seen that the distance from the cylinder centre to the bottom of the flanges may be as small as possible, consistent with the necessary clearance between the covers and the bed plate. The height of the cylinder centre above the bed plate should always be as small as possible, as the bed is recessed to give clearance for the crank pin.

(155) **Length of Cylinder.**—The length of the cylinder inside from cover to cover is evidently decided by the length of the piston stroke, the width of the piston, and the clearance

between the piston and covers when at the end of the stroke. But as the covers are always let into the cylinder for a short distance, as shown in Figs. 180 and 181, the total length of cylinder from face to face, outside, and without covers, is equal to—

(Stroke of piston) + (width of piston) + (clearance × 2) + (distance the covers are let into cylinder × 2).

If the cylinder has only one loose cover, as in Fig. 180, the length will be less by the distance for one cover.

(156) **Clearance.**—This is made as small as possible consistent with safety, as all clearance means a waste of steam. It should not be less than $\frac{1}{8}''$, and for small sized engines is usually $\frac{3}{16}''$ or $\frac{1}{4}''$. For vertical engines the clearance is greater at the bottom, and should be greater for high than for low speeds.

(157) **Size of Steam Ports.**—The amount of steam which can enter the cylinder depends upon the size of the steam port, and as the steam required depends upon the piston speed, it is evident that the port areas must bear some relation to this speed. It is usual to proportion the ports so that the velocity at which the exhaust steam must pass through them in order to escape freely in front of the piston is not greater than 100 feet per second, but when convenient it is better to make the ports large so that the velocity need not exceed 80 or 90 feet per second, the former being a usual practice. Hence, if we know the piston speed and diameter, we can determine the port area, for we have—

$$\text{Area of steam ports in square inches} = \begin{cases} \dfrac{\text{Volume swept through by piston per second}}{\text{Velocity of steam per second}} \\ \text{That is—} \\ \dfrac{\text{Area of piston in square inches} \times \text{piston speed in inches per second}}{\text{Velocity of steam in inches per second}} \end{cases}$$

These proportions give ample size for the steam to get into the cylinder, since its pressure is then greater than at exhaust, although, as will be seen later, the whole of the port is not uncovered for the admission.

The larger the port in a direction at right angles to the movement of the valve, the larger the area uncovered for the same movement, therefore the aim is to get the ports as long and as narrow as possible. Practical experience has adopted a port length varying from 0·5 to 0·75 of the cylinder diameter, the most common length being $0{\cdot}6d_c$. Hence the width of the port can be found by dividing the port area by the port length, taking dimensions for the length to the nearest $\frac{1}{4}''$, and for the width to the nearest $\frac{1}{16}''$.

(158) **Exhaust Port and Bars.**—The conditions which fix the width of the exhaust port and of the division bars between the ports are, that a wide port and bars give more room for the exhaust passage, and shorten the two steam passages, thus diminishing the clearance, but require a larger slide valve, thus increasing the friction between the valve and port face, and thereby the work done in moving the valve, and also the size of the eccentric and valve rods. The width of the exhaust port must not be less than will give a width of opening to the exhaust passage of not less than the steam port width when fully open to exhaust,* and in small engines is usually made equal to twice the steam port width, the width of the bars being equal to the steam port width. It is necessary for the bars to be at least $\frac{1}{8}''$ to $\frac{3}{16}''$ wider than the thickness of the walls of the steam passages, in order to afford a convenient edge for working the ports true —this will be seen in the ports of Fig. 181—and it should be noticed that the same thing occurs at the other edges of the steam ports, so that the steam passages are always wider by some $\frac{1}{8}''$ than the ports.

In order to diminish clearance the steam passages are made to enter the cylinder as near the ends as possible, and the covers are let in so as to just reach the edge of the passages. This is seen in both the examples, while in Fig. 180 it is shown how the bottom and top covers are scooped out at the steam passages in order to obtain a better opening when the piston is at the end of its stroke. All corners and bends in steam passages should be well rounded, in order to reduce resistance to the flow of the steam.

(159) **Steam and Exhaust Pipes.**—The diameters of the openings for the steam and exhaust pipes are also determined by the consideration that the size must be sufficient to allow free passage of the steam without requiring too high a velocity. A small steam pipe means loss of pressure, and a small exhaust pipe a slow exhaust. If the steam pipe is so arranged that the velocity shall not exceed 100 feet per second, there should be no loss of pressure, but practical results show that larger inlets and outlets are preferred. The steam and exhaust pipes have always a greater area than the steam and exhaust ports, so that the velocity does not exceed 90 feet per second for steam, and 70 feet per second for exhaust. Average values are 70 feet for steam pipes, and 50 feet for exhaust pipes, on such engines as we are considering. The area of the pipes are found exactly as the

* It will be of great assistance to read §§ 171 to 178 on the design of slide valves before commencing to design the exhaust ports, as the reason for this is there explained.

steam ports, and from the areas the diameters can be obtained. Diameter of pipes dimensioned to nearest $\frac{1}{8}''$.

(160) **Thickness of Metal.**—It has already been pointed out that the thickness of metal in castings is considerably in excess of what is required for strength, owing to the need for uniformity in cooling and for soundness in casting. Thisis true for cylinders, and as a result the thickness of small cylinders always seems excessive, the minimum practical thickness being $\frac{7}{16}''$ or $\frac{1}{2}''$. When a liner is not fitted, the barrel should have an excess in thickness to allow for reboring, and also the port face to allow for refacing. The thickness of the covers depends upon their shape, and whether they are provided with strengthening ribs, as in the steam chest cover of Fig. 181, or with projecting bosses or flanges for stuffing boxes; a simple flat plate being a weak section, and one to be avoided.

For cylinders up to 10″ diameter, the thickness of the barrel should be from $\frac{1}{2}''$ to $\frac{3}{4}''$, without liners. With liners the thickness may be less by $\frac{1}{8}''$ (to a minimum of $\frac{1}{2}''$), the liner being $\frac{5}{8}''$ to $\frac{11}{16}''$ thick. The thickness of the walls of the steam passages and of the valve chest sides may be from $\frac{1}{16}''$ to $\frac{1}{8}''$ less, thus being $\frac{7}{16}''$ to $\frac{9}{16}''$ thick. The flanges for the covers should be treated in the same way as already shown in §§ 44, 47, the flange thickness being equal to $1\frac{1}{4}$ to $1\frac{3}{8}$ times the thickness of the barrel-cylinder, and steam chest covers should be fitted with studs, which should be arranged, if possible, so as not to enter the steam ports.

(161) **Stuffing Boxes.**—These should be designed according to the directions and rules of Section xxiv. Notice that the thickness of the metal of the stuffing box need not be as great as the cylinder walls, it may be equal to the thickness of the cylinder walls, less the amount allowed for reboring. The flange thickness should be sufficient to give enough length for the screwed part of the gland studs.

(162) **Boring of Cylinders, Lagging, Drain Cocks, Indicator Bosses.**—The cylinder barrel must be bored slightly larger ($\frac{1}{8}''$ to $\frac{1}{4}''$) than the piston diameter at each end, and for such a distance that the piston and the nearest edge of the piston ring (but not the whole of the width of the ring, or it will fly out of the piston) shall overlap the smaller diameter length when at the stroke end. This is clearly shown in Fig. 181, the reason being to assist in getting the piston and rings into place, and to prevent a *burr* being formed by the wear of the cylinder walls.

For the same reason the port face must be of such a length that the slide valve shall overlap it when at the end of its stroke (see § 173).

It will be noticed that the cylinder and slide chest cover of Fig. 181 have projections outside the flanges. These are simply to provide a convenient means of fixing the lagging with which cylinders should always be covered.

Arrangements must be made for draining the cylinder (in a horizontal engine from each end) and for providing for the lubrication of the piston and valves by an oil supply to the steam chests. This simply means that a boss must be provided in the position of the drain cocks and lubricators. Engine makers are now making provision for the fitting of indicators on the cylinders. This is done by casting bosses, one at each end, so that the holes for the indicator cocks lead into the clearance space between the piston and cylinder cover when at the end of the stroke. These are shown in Fig. 180. Indicator cocks are screwed to gas thread, and the height of the boss should extend beyond the lagging.

(163) **Facings for Machining.**—In designing any casting, such as a cylinder, condenser, crosshead, bed plate, &c., care must be taken that all parts which require machining shall project by some $\frac{1}{8}''$ to $\frac{1}{4}''$ beyond the metal surrounding them. This applies to all cases of covers, flanges, bosses, &c., examples of which will be seen in the condenser of Fig. 197. All covers should be let into the parts they cover, as this prevents side movement independently of the studs. There are other, apparently small, but really important matters, such as these which need attention but which are too numerous for mention here. It should be the aim of the student to carefully understand the reasons for the form and position of every part he draws, however apparently unimportant, and if doing so he is able to produce a better design, then he should not be tied to the exact details of any copy.

(164) **Order of Drawing.**—The following will be a convenient order to follow in designing and drawing a cylinder, having reference to such a drawing as Fig. 180, A :—

(1) Centre lines of piston and valve-rods, and centre lines through middle of ports and cylinder (marked *a b*, *c d*, and *e f* respectively); (2) lines showing diameter of cylinder and diameter outside liner in valve chest; (3) draw outline of piston at top of stroke, with piston-rod, nut, and washer (distance from centre line to middle of width of piston rim = $\frac{1}{2}$ stroke); (4) allow for clearance and draw outline of inside of cover, and then outside of cover knowing thickness; (5) draw outline of piston at bottom of stroke and do the same for the bottom end of the cylinder; (6) set off port openings at the liner and draw in steam passages, allowing piston to overlap passage the given

distance; (7) draw in outer wall of steam chest and the division plates; (8) draw in liner in both elevation and plan, and show ports (length of liner to allow of valve overlapping at limits of stroke); (9) complete top cover of cylinder, and top and bottom covers of steam chest; (10) complete bottom end of cylinder and stuffing box; (11) finish details. It is very convenient to show the piston and slide valve at least in correct outline in the cylinder drawing—this should be done before the drawing is inked in.

EXAMPLES.

EX. A 1.—Make working drawings, fully dimensioned, showing sectional plan and front elevation, end elevation and plan not in section, to scale of half full size of the vertical cylinder of Fig. 180. Diameter $7\frac{1}{2}''$, stroke $5\frac{3}{4}''$, piston rim $1\frac{3}{4}''$ wide, rod $1\frac{1}{4}''$ diameter, $1\frac{1}{4}''$ hex. nut, with washer and keep pin; slope of under side of piston and inside of bottom end of cylinder 60°, with centre line of rod; top clearance $\frac{3}{16}''$, bottom $\frac{1}{4}''$, top cover let in $\frac{7}{8}''$. Centre of cylinder to centre of valve $9\frac{1}{4}''$, valve liner $4''$, outside diameter $\frac{1}{2}''$ thick, $6\frac{5}{8}''$ long; upper and lower ports $\frac{5}{8}''$ wide, and to have an area for a piston speed of 400 feet per minute, and a steam velocity of 85 feet per second; centre port of same length but $1\frac{5}{8}''$ wide, bars between ports in liner $\frac{7}{8}''$; steam passages to cylinder $\frac{3}{4}''$ wide, thickness of division plates in steam chest $\frac{5}{8}''$. Steam pipe diameter for a steam velocity of 85 feet per second, exhaust pipe for 75 feet velocity. Valve-rod $\frac{7}{8}''$ diameter at stuffing box, least thickness of top cover $\frac{1}{2}''$, of bottom cover $1\frac{1}{8}''$. Metal of cylinder walls $\frac{5}{8}''$, of steam passage walls $\frac{7}{16}''$, of valve chest and cylinder cover $\frac{1}{2}''$, of bottom end of cylinder $\frac{5}{8}''$ to $\frac{7}{8}''$, of stuffing box $\frac{1}{2}''$, flanges of cylinder $1''$, flange of cylinder cover $\frac{7}{8}''$, diameter of bottom flange $12\frac{1}{4}''$, flanges for steam and exhaust $\frac{3}{4}''$, other sizes as shown. Show piston and valve-rod glands. (Details of piston from Ex. A 2, p. 273.)

(Notice that cylinder at top and bottom, near steam passages, including top cover, is scooped out to allow the steam free entry into the clearance space. Size marked A in plan must be equal to sum of length of ports, in order that area of passage may equal area of port. Clearance between piston and cylinder ends to be in direction of motion, not perpendicular to face of piston; clearance around piston not to be less than shown in figure. Cylinder cover need not fit cylinder for whole of distance it is let in. Covers of cylinder and valve chest are shown bedding on two faces, when this is done the joint is really made at the inner face, but the usual construction of Fig. 181 may be adopted. Holes for covers of valve chest to be of slightly larger diameter than largest diameter of liner, the holes are circular, but the flanges on the covers are oblong, with semicircular ends, the studs for fixing to chest just missing the centre line, and, therefore, not seen in the sectional elevation. Cylinder walls to be thickened for length slightly

greater than part turned of larger diameter at two ends. Owing to piston-rod and crosshead being in one piece, the least diameter of stuffing box, without bush, must not be less than 2″, the largest diameter of the piston-rod and the gland and back bush must be split, the gland being held together by a steel washer.)

EX. B 1. — Make working dimensioned drawings showing three views, scale, half full size, of a horizontal steam-jacketed cylinder, as in Fig. 180. Diameter, $8\frac{1}{2}''$; stroke, 12″; height of centre above bed plate, 8″; piston, $2\frac{3}{4}''$ wide; rod, $1\frac{1}{4}''$ diameter, increasing to $1\frac{1}{2}''$ at screwed part; hexagonal nut with washer, clearance each end, $\frac{3}{16}''$; covers let in $1\frac{1}{16}''$ each end; liner, $\frac{5}{8}''$ thick, increased to $\frac{3}{4}''$ for length of $2\frac{3}{4}''$ at each end, where fitted to cylinder; jacket space, $\frac{3}{4}''$; walls of steam passage, $\frac{1}{2}''$. Centre of cylinder to port face, $7\frac{7}{8}''$; to centre of main slide valve, $8\frac{13}{16}''$; to outside of steam-chest flange, $13\frac{7}{8}$; centres of valve-rods, $2\frac{1}{4}''$ apart; rods, $\frac{3}{4}''$ diameter. Steam chest—outside, $17\frac{1}{4}''$ long, $8\frac{1}{4}''$ wide. Thickness of cylinder flanges, $\frac{7}{8}''$; outside diameter at ends of cylinder, $13\frac{3}{4}''$; diameter of projecting flange for lagging, $15\frac{1}{2}''$; metal of cylinder covers, $\frac{5}{8}''$ thick; of steam chest cover, $\frac{3}{4}''$; of sides of steam chest, $\frac{9}{16}''$. Steam ports to have an area for a piston speed of 300 feet per minute, and a steam velocity of 90 feet per second; steam pipe area for a velocity of 50 feet per second, exhaust pipe for 30 feet velocity; from centre of cylinder (in plan) to centre of steam inlet, $7\frac{7}{16}''$; to centre of exhaust outlet, $6\frac{1}{4}''$; steam-inlet centre, $5\frac{1}{8}''$ from centre line through ports; exhaust-outlet centre, $1\frac{1}{8}''$. Bosses for cylinder drains, $1\frac{3}{4}''$ diameter. Other sizes as in Fig. 181. Details of piston, Fig. 182, *b*, and Ex. B 2, p. 273.

(Notice from the plan, that in order to obtain a sufficiently large exhaust passage, E P, it is necessary to lessen the steam jacket space at that part; it would be better if the steam passages could be less curved than shown. around the exhaust passage; the steam passages are to lead into cylinder, close up to covers. Stuffing box in front cover much longer than usual, in consequence of lugs for guide bars, design boxes for valve-rods of usual depth. Outside ribs on steam chest cover are convenient to fix a plate across, the inside space being filled with non-conducting material. Total length of port face not to exceed $6\frac{3}{8}''$, in order that valve may overlap at ends of stroke. The feet for bolting down to bed plate are connected across the cylinder ends by a web $\frac{1}{2}''$ thick, the bracket marked B being cast at each end, having a bottom flange about 3″ wide, not bolted down, for extra supports.)

(165) **Pistons.**—The chief points to be secured in a steam engine piston are that it shall be as light as possible consistent with rigidity in order to reduce its inertia, that it shall occupy a small space, not taking up too much of the length of the cylinder, and that it shall be steam-tight against the cylinder walls without exerting too great a pressure, in order to prevent

the leakage of steam past the piston, and to reduce the frictional resistance to movement to a minimum.

(166) **Piston Rings.**—Pistons are made steam-tight by inserting spring rings, made of cast iron, steel, or hard bronze, into grooves turned in the piston rims, the rings being made to a diameter slightly larger than the cylinder, so that when sprung into position they shall exert a pressure against the cylinder walls. The theory of piston rings is that they can be made of such an unequal thickness, or if of equal thickness they can be bent to such a curve (which is not an exact circle) so that when in position they shall exert an equal radial pressure, the magnitude of which per square inch depends upon the amount of the inequality in thickness, or upon the form and size of the curve to which the ring is bent. The width of the ring in the direction of the cylinder length does not affect the radial pressure per square inch, hence there is no reason why a wide ring should be better than a narrow one if the rubbing surface is a good one. It has been found that a radial pressure of $3\frac{1}{2}$ to 4 lbs per square inch is sufficient to ensure a tight piston against a steam pressure of 100 lbs. per square inch. A wide ring exerts a greater total pressure against the cylinder than a narrow one, and, therefore, increases the frictional resistance to movement; but it gives more guiding surface than a narrow ring, and may possibly be preferred for that reason. Two rings are better than one, as the joints of each can then be arranged at different positions of the piston; in such cases the rings must be prevented from turning round. The piston is often turned slightly less in diameter than the cylinder ($\frac{1}{16}''$ to $\frac{1}{8}''$) so that only the rings rub against the walls.

Practical customs differ very widely in the width of rings to be used; some makers always fit narrow rings, others wide ones. The well-known and commonly used Ramsbottom rings have a width of about $\frac{1}{50}$ the piston diameter, the least width being $\frac{3}{8}''$, whereas cast-iron single rings may have a width of $\frac{1}{5}$ the piston diameter. It is well to remember that a wide ring usually means a wide piston, for we simply proportion the piston rim to surround and support the rings. A very important point is that the ring must not lap over the steam passage when at the end of its stroke for its whole width, otherwise it may spring out and catch the edge of the metal, nor must the ring come wholly into that part of the cylinder end which is turned of larger diameter. There are a large number of useful patent piston rings and springs in use, but chiefly in cylinders of large diameter.

(167) **Forms of Pistons.**—Three different forms of pistons are shown in Fig. 182, *a* and *b*, and in the cylinder of Fig. 180. The piston of Fig. 182, *b*, is known as the "box pattern," and is

made of cast iron with one loose plate P, the more common custom being to cast the piston in one piece, leaving holes in the front and back plates for removing the core used in casting, which are afterwards plugged up. There are two bronze rings placed in contact, the ends being cut diagonally, as shown, without lapping, and an inner steel ring equal in width to the two bronze rings. The pistons of Figs. 180 and 182, *a*, are distinguished as "single thickness pistons," and are made of cast steel, their advantage being that they are much stronger for the same weight than the box pattern, and are more easily cast. Their disadvantage is that at least one cylinder cover must agree in shape with the section of the piston when as in Fig. 182, *a*, and both covers

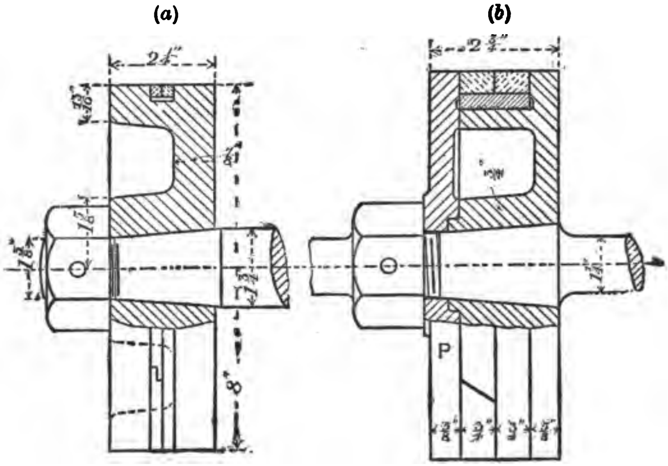


Fig. 182.

when as in Fig. 180, in order to avoid a large clearance space. This is clearly seen in Fig. 180, where the piston is shown at the end of its stroke, the top cover and the bottom end of the cylinder having the shape of the corresponding face of the piston. This piston is known as the "conical" or "dished" type, the reason for such a section being that it gives the same strength for a less thickness of metal than when made as a flat plate as in Fig. 182, *a*, while another advantage is that it gives a greater length from the shaft centre to the cylinder gland, for the same length of standard. All the best designed pistons of large diameter are now made of this pattern. Each of the pistons in Figs. 180 and 182, *a*, has two steel rings; those in the former are in contact,

each being $\frac{1}{4}'' \times \frac{1}{4}''$, while in the latter the rings are $\frac{7}{16}''$ wide $\times$ $\frac{3}{32}''$ thick, placed in grooves $\frac{7}{16}''$ apart. The bronze rings of Fig. 182, *b*, are $\frac{3}{4}''$ wide $\times$ $\frac{1}{2}''$ thick, and the inner steel rings $1\frac{1}{2}''$ wide $\times$ $\frac{7}{16}''$ thick. Narrow rings should be half side lapped where the ends meet as shown in Fig. 182, *a*. The proportions of the centre boss are decided by the piston-rod.

(168) **Piston-Rods.**—Piston-rods are alternately subject to tensile and compressive stresses, and should be designed so that the smallest section, such as at the bottom of the thread, or through the cotter way when so fixed to the crosshead, is safely able to stand a tensile and compressive stress, caused not only by the steam pressure upon the piston, but also by the inertia of the moving parts connected to the rod, as the latter are too large, especially in high-speed engines, to be neglected. But owing to the length of the rod, it ought also to be designed as a strut, in order to prevent the possibility of buckling. Such considerations are, however, beyond the limits of this book. For such engines as we are dealing with, these effects will be sufficiently allowed for if the rod is made of such a diameter at the smallest part, as that the stress produced by the maximum steam pressure alone, does not exceed 3,500 lbs. per square inch.

(169) **Fixing Piston to Rod.**—The usual method of fixing the piston-rod to the piston is by enlarging its end, and then turning it taper for a length, equal to the length of the boss, leaving a screwed part beyond, for a nut, which, being screwed home, holds the piston tightly in its place. The three pistons illustrated are fixed in this way, and great care must be taken to fix the nut so that it cannot work loose; the most common method being to drive a taper-split pin through the nut and rod (see Fig. 182). When the taper part is long compared with the diameter—say, $1\frac{1}{2}\,d$ to $2\,d$, as in Fig. 183—the *total taper* may be from 1 in 5 to 1 in 9, but when the taper part is less than this, as in Fig. 180, the total taper may be as much as 1 in $1\frac{1}{4}$. The thickness of the nut is often made slightly greater than the ordinary standard, and is screwed up against a washer from $\frac{3}{8}''$ to $\frac{1}{2}''$ thick.

The design of the crosshead end of the rod will be considered in the section on crossheads.

(170) **Drawing of Pistons.**—In making working drawings of pistons, it is convenient to show two views, one looking on the piston face to which the nut is screwed; the other, as in Fig. 182, the greater half of the view being in section. The piston-rod, for a short length, and its manner of fixing to the piston should be shown. Notice that the thread must extend for a short distance within the piston, so that the nut may not reach the end of the thread before the rod is screwed tightly home. It is well to notice that the position of the spring rings must be settled before complet-

ing the cylinder drawing, otherwise it is not possible to decide what length of cylinder must be bored, of larger diameter at each end, so that only part of the ring may overlap. There must be clearance between the bottom of the rings and the grooves in which they are placed.

EXAMPLES.

EX. A 2.—Make working drawings of a steel-dished piston for the cylinder of Fig. 180. Full size. Rod, $1\frac{1}{4}''$ diameter, screwed part, $1\frac{1}{4}''$; taper, 1 in $1\frac{1}{8}$; thickness of piston at boss, $1''$, tapering from $\frac{9}{16}''$ to $\frac{5}{16}''$ at rim; rim, $1\frac{3}{4}''$ wide; thickness, $\frac{1}{2}''$ to $\frac{3}{8}''$; two spring rings, $\frac{7}{16}'' \times \frac{3}{16}''$; nearest edge of rings, $\frac{1}{4}''$ within edge of rim. Bottom face of piston at 60° to centre line of rod. Other dimensions as in Fig. 180.

EX. B 2.—Make working drawings of a cast-iron box piston with one loose cover, as in Fig. 182, *b*, for the cylinder of Fig. 180. Full size. Rod, $1\frac{1}{4}''$ diameter, passing through back and front of cylinder; screwed part, $1\frac{1}{2}''$ diameter; taper, 1 in 9. Width of piston, $2\frac{3}{4}''$; thickness of plates, $\frac{5}{8}''$. Two bronze spring rings, $\frac{3}{4}''$ wide, $\frac{1}{2}''$ thick; one inner steel ring $1\frac{1}{2}''$ wide, $\frac{7}{16}''$ thick.

(Notice the small projection on the piston boss, and the corresponding groove in the plate, also the projection on the plate to form a face for the nut, and the recess in the inner face of the plate to save turning.)

(171) **Slide Valves.**—The most common form of slide valve which, owing to its cross section resembling the letter D, is known as the D-slide valve, is shown in Fig. 183, a cross-section

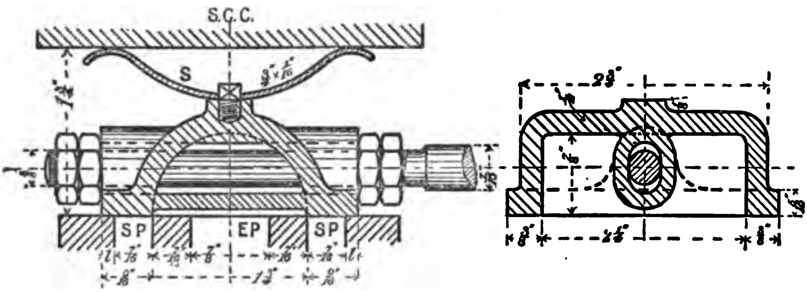


Fig. 183.

of the valve through the centre being shown in the separate figure on the right-hand side, the width inside the valve in this section being equal to the length of the ports. The valve is shown in position at mid-stroke, and it will be seen that when it is moved either to the right hand or to the left, steam will

enter one of the ports, S P, from the space outside the valve, while the exhaust steam from the cylinder will pass out of the other port into the inside of the valve, and then into the exhaust passage, E P. The round boss on the two ends of the valve extends through the valve, and surrounds the valve-rod, which is fixed by lock nuts as shown. The hole through these bosses should be elongated, as seen in the cross-section, in order to allow of a little play between the valve and the nuts on the rods, so that the valve may be kept hard against the port face by the steam pressure upon its back. The flat spring, S, fixed by a square-ended set screw and working against the steam chest cover, S C C, also serves to keep the valve in place. The sizes given are for a horizontal cylinder 4″ diameter, 6″ stroke.

(172) **Proportions of Valves, Lap, and Lead.**—The methods by which the right proportions of a slide valve and its attached parts are obtained, in order to ensure a required distribution of the steam, cannot be explained here, as the limits of the book will not permit of an explanation of the principle and use of valve diagrams, without which such problems are not capable of graphical solution. Students who possess sufficient knowledge of the working of slide valves will find the question of valve diagrams very fully treated in Professor Unwin's *Machine Design*, Part II.

In such engines as we are considering, steam is usually cut off for expansion at five-eighths of the stroke, and for compression at from nine-tenths for low speeds, to four-fifths for high speed, the release to exhaust being not earlier than nine-tenths the stroke, the valves having a lead of $\frac{1}{8}''$ to $\frac{3}{16}''$. Vertical engines should have a greater bottom than top lead. For a valve to give these results, it must have both steam and exhaust lap—that is, when the valve is in mid-stroke, it must overlap the steam port on both sides, and the eccentric sheaves must be so fixed upon the crank shaft as to give an opening of the port to steam, equal to the given lead, when the piston is at the end of its stroke. The valve of Fig. 183 has only a steam lap, equal to the amount marked *l*.

(173) **Travel of Valve and Width of Exhaust Port.**—It will be seen from Fig. 183, that before the left-hand port can be fully open to steam, the valve must move to the right, a distance equal to the width of the steam port + amount of steam lap; and, also, that before the right-hand port can be similarly open, the valve must move a like distance to the left of its mid-position. Hence the "*travel*" or "*stroke*" of the valve during one stroke of the piston is as follows:—

Travel of valve = 2 (width of steam port + steam lap).

When the steam port is not fully uncovered for the admission of steam, this becomes—

Travel of valve = 2 (width of port opening to steam + steam lap).

We see from this that the length of port facing beyond the outer edge of the steam port for valves, as in Fig. 183, must be less than (width of steam port + steam lap), in order that the edge of the valve may overlap the facing when at end of stroke.

When the valve is in its extreme right- or left-hand position, the exhaust port must be uncovered for a width *at least* equal to the steam port width. In order that this may be for the valve of Fig. 183, with no inside or exhaust lap, it is evident that—

Width of exhaust port = width of steam port + steam lap.

When the valve has exhaust lap, then the least width of the exhaust port must be—

Width of exhaust port = width of steam port + steam lap + exhaust lap.

It has already been pointed out that the exhaust port width is usually in excess of this amount.

(174) **Pressure on Valves.**—It is evident that valves of the type of Fig. 183, having faces in contact with the port face, and exposed to the steam pressure within the valve chest, must offer considerable resistance to movement caused by the friction between the faces in contact. This frictional resistance is indeed the only force which acts against the movement of the valve (neglecting the friction of the stuffing boxes), and, therefore, produces a stress in the valve-rod, eccentric-rod, and eccentrics. Some experiments made by Mr. J. A. F. Aspinall, M. Inst. C.E. (*Inst. of C.E. Proceedings*, vol. xcv.), showed that the coefficient of friction between the valve and port faces for locomotive slide valves varied from 0·05 to 0·068, but that even with so low a coefficient, the work done in driving the valves was as much as 2·4 H.P. per valve, or 1·13 per cent. of the total I.H.P. of the engine. In designing the valve-rod and connected parts, it is better to take a value for the coefficient of friction, "μ," not less than 0·2 or 0·25, in order to allow for the extra force required to start the valves, and for the friction of the stuffing box. In finding the total pressure producing friction, allowance must be made for the fact that some parts of the valves may be subject to pressure acting away from the port face, such as when a steam port is partly or wholly covered by the valve; it is usual to consider that one steam port and the exhaust port is covered, and, therefore, if P be the total pressure producing friction, A the area of the back of the valve, p the initial steam pressure, p_1 the

the exhaust steam pressure, a the steam port area, and a_1 the exhaust port area, then :—

$$P = p\ (A - a) - p_1\, a_1.$$

In a non-condensing engine where p is taken above the atmosphere, then the value of $p_1\, a_1$ is neglected, and the $P = p\ (A - a)$. Also, if F be the frictional resistance to movement, to be overcome by the eccentric and rods, and μ the coefficient of friction, then $F = \mu\ P = \mu\ p\ (A - a)$.

(175) **Piston Valve.**—The chief object in using a piston valve as in Ex. A, in preference to the ordinary flat D valve, is that no parts of it in contact with the port face are pressed upon by the

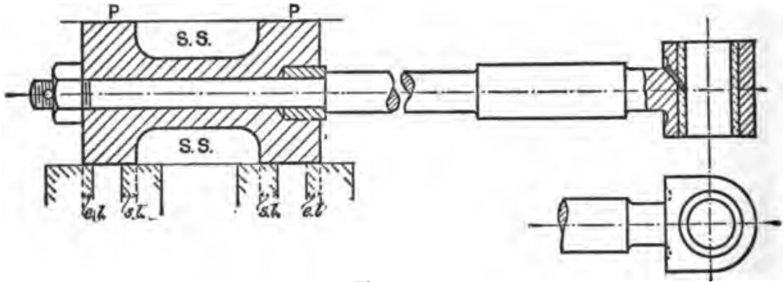


Fig. 184.

steam, and, therefore, the above resistance to movement is almost entirely avoided. As will be seen from Fig. 184 the valve is a simple cylindrical casting, the centre part of which is turned to a smaller diameter than the ends. The valve is shown in the

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Fig. 185.—INDEX TO PARTS.

(*a.*)—Sectional Elevation of Valves; left-hand half of main valve, M V, is a section through centre line, and the right-hand half a section through the line *s t*.

(*b.*)—Plan of Top Face of Main Valve.

(*c.*)—Cross Section of Valve through centre line, *x y*.

(*d.*)—Front Elevation and Plan of Expansion Valves, E V, with nut for screwed part of rod.

E.V. for Expansion valve.
N. „ Expansion valve nuts.
M.V. „ Main valve.
p. „ Passages through main valve for steam.
s.s. „ Exhaust steam space in main value.
b. for Bosses for surrounding valve-rod.
s.l. „ Steam lap.
e.l. „ Exhaust lap.
G.R. „ Guide-rod for main slide valve.
E.G.R. „ Guide-rod for expansion valve.

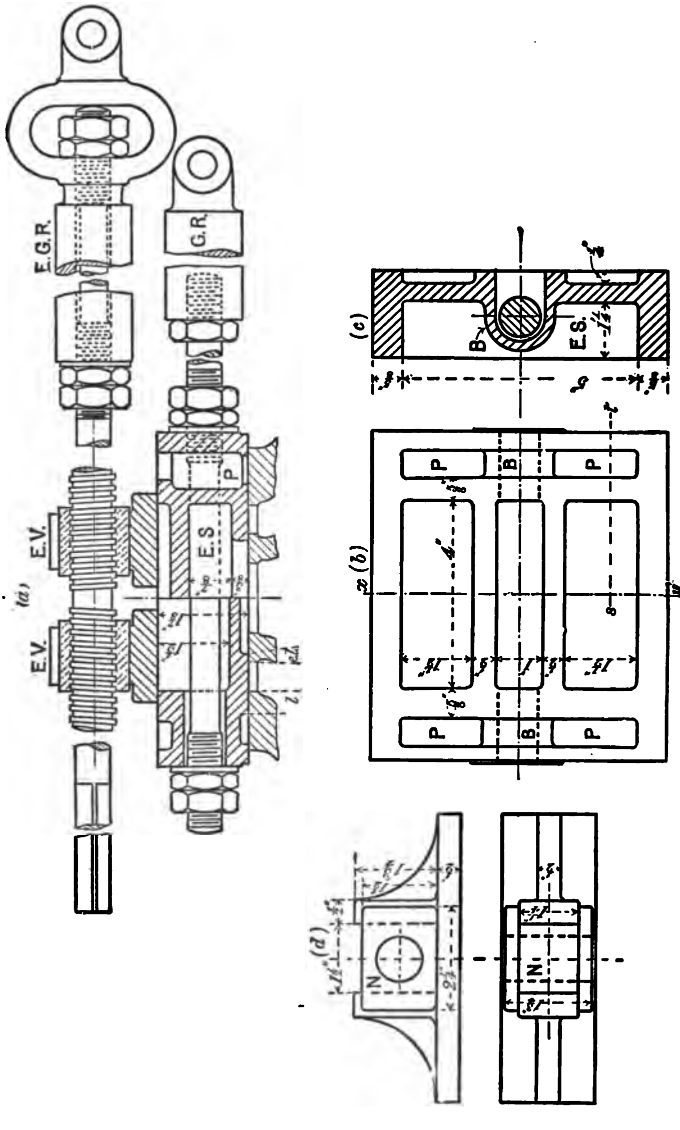


Fig. 185.

figure at half stroke, the initial steam surrounding the part marked S S, and being, therefore, *inside* the valve, while the exhaust steam is above and below the valve, as already described in connection with the cylinder drawing, Fig. 180; *s l* is, therefore, the steam lap and *e l* the exhaust lap, steam being cut off by the inner edges of the valve. The rubbing or piston parts of the valve P P are not packed with rings, but simply turned to a good working fit; but with larger sized valves of this pattern, spring rings are used exactly as with ordinary pistons. The valve-rod passes right through the centre of the valve as shown, the enlarged part being for the valve-rod guide, which will be afterwards described, as well as the connection of the rod to the eccentric-rod.

(176) **Main and Expansion Valves.**—In Fig. 185 (*a*), (*b*), (*c*), (*d*), are shown the valves for the horizontal engine of Ex. A. The lower or main valve, M V, is provided with two steam passages, P P, leading from back to front, the centre space, E S, being for

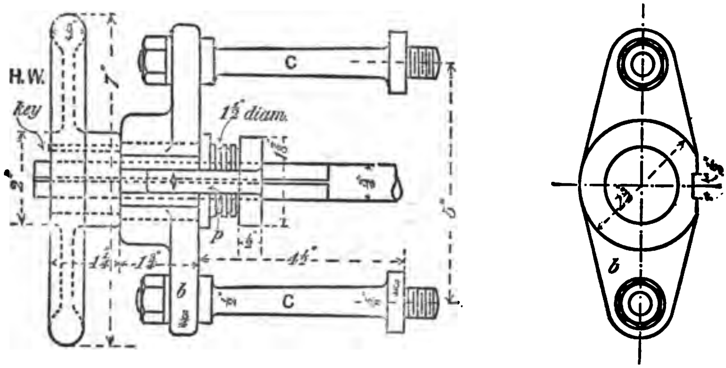


Fig. 186.

the exhaust steam. The valve-rod passes through the whole length of the valve, is fixed with lock nuts at either end, and is surrounded by bosses, B B, where it passes through the steam passages. On the back of this valve there works two other valves, E V, with simple flat faces and of a construction shown in the figure, both threaded on one rod which is provided with a right- and left-handed square thread screw, so that when the rod is turned round the valves separate or approach equally, and thus alter the point at which steam shall be cut off. The valves are fitted with gunmetal flanged nuts, N, screwed to receive the rod. The back end of the rod for these valves passes through a stuffing box in the back of the steam chest (see Fig. 181),

and is then attached to the expansion gear shown in Fig. 186. The turned columns, C C, which support the crossbar *b*, are screwed into the flange of the stuffing box cover. Inside the crossbar is fitted a flanged brass bush, screwed internally to receive a second bush having a projecting piece, *p*, fitting into a groove cut in the front edge of the crossbar, and graduated to show the point of cut-off. The valve-rod end is made square for some distance, and fits a square hole through the outer bush, so that when this bush is turned round by the hand-wheel, H W, which is fixed to it, the valve-rod is turned, and the inner bush, having a round hole, is moved to and fro, and marks the position of the expansion valve. Dimensions for this gear are marked on the figure, and it should be regarded as a detail to be shown on the detail drawing of the valves. The inner or main valve affects the points at which release to the exhaust and cut-off for compression shall take place, and acts in this way just as an ordinary valve.

(177) **Valve-Rods.**—Details of the valve-rods will be considered with the eccentrics, as it is sufficient in the valve drawings to show only that part of the rod attached to the valve, but it should be noticed here that their diameters must be sufficient to stand the stress produced by the frictional resistance of the valves, in addition to which they must be strong enough to resist buckling. For small engines, such as we are dealing with, it will be found that the least diameter practically convenient is generally more than sufficient for strength, especially with valves of the piston type. But care should be taken that the rod is not subject to a greater stress than 3,500 lbs. per square inch on the smallest section, due to the frictional resistance alone.

(178) **Size of Valves.**—There can evidently be no object in having larger valves than absolutely necessary ; the valve length is, of course, settled by the port face and the lap, but in width the valves need not exceed the port length by an amount at each side greater than 1 to $1\frac{1}{4}$ times the width of the bars between the steam and exhaust ports. The height of the valve from the port face is partly a question of convenience in designing the cylinder, and should be as little as possible. It is unnecessary to consider any rules about thickness of metal, from the fact of their being castings, the thickness cannot be less than $\frac{3}{8}''$ to $\frac{1}{2}''$, and is thus of ample strength. It should also be noticed that the steam chest cover is provided with inside ribs which do not actually touch the slide valves, but serve to prevent them moving too far from the port face, and that the length of the steam chest must evidently be sufficient to give clearance when the valves are at the end of their stroke.

EXAMPLES.

EX. A 3.—Make working dimensioned drawings full size, showing two views of a piston slide valve for Ex. A, as in Fig. 184. Outside exhaust lap, $\frac{3}{16}''$; inside steam lap, $\frac{7}{16}''$; diameter of valve at small part, $1\frac{1}{2}''$; rod, $\frac{5}{8}''$ through valve; $\frac{7}{8}''$ beyond valve. Diameter of valve and size of ports from cylinder drawing, Fig. 180.

(Find distance between centre of steam ports from Fig. 180, and draw outline of ports, then width of piston part of valve = width of port + outside lap + inside lap. It is better to screw the collar on the rod where it enters the valve against a wrought-iron or brass bush, rather than against the cast-iron valve, therefore show such a bush, and the way the rod is fixed; show a keep pin to prevent the nut working loose.)

EX. B 3.—Make working dimensioned drawings full size, showing three views of the expansion and main slide valves for Ex. B, as in Fig. 185. Expansion valve faces, $6\frac{1}{4}''$ long, $1\frac{7}{8}''$ wide. Main valve—total length, $7''$; total width, $6\frac{1}{4}''$; total depth, $\frac{7}{8}''$; steam lap, $\frac{1}{2}''$; exhaust lap, $\frac{1}{8}''$; width of openings through valve, $\frac{5}{8}''$; rod through main valve, $\frac{3}{4}''$ diameter; expansion valve rod, $\frac{3}{4}''$ diameter; $1''$ diameter square thread screw for length of, $7''$. Length of gunmetal nuts in expansion valves, $1\frac{3}{4}''$. Show the expansion arrangement of Fig. 186 on the same drawing.

(Make outline drawing of port face and back end of steam chest, with the flange for the expansion gear, and showing centre lines of valve-rods, sizes from Fig. 181 and Ex. B1. Notice that passages through main valve are widened out so that only the metal near the top and bottom face needs machining, also that top face of valve is recessed to save machining; show projecting faces on ends of main valve for nuts to bed against, and turn valve-rod down between the screwed ports so that it may be easily passed through the valve. Lock nuts at each end. The gunmetal nuts for the expansion valves are dropped in place from above, and bear against the flanges, one of which is shown at each side.)

SECTION XXVIII.

CROSSHEADS—GUIDES—CONNECTING-RODS.

THE crosshead, guide blocks, and the crosshead or small end of the connecting-rod all connect at the same part, and as the form of each depends so much upon the arrangement of the whole, it will be convenient to consider them together before dealing with the details of the separate parts.

(179) **Crosshead.**—The crosshead is the name given to the piece fixed to the outside end of the piston-rod, and moving with it. The "*guide block*" or "*slide block*" is that part which is fixed to the crosshead, or forms a part of it, and moves with it, between parallel parts called "*guides.*" The "*crosshead pin or gudgeon*" is the part attached to the "*connecting-rod,*" which rod joins the crosshead to the crank pin.

(180) **Forms of Crossheads.**—The crosshead may be in the form of a block rigidly fixed to the piston-rod, or forming a part of it, containing brasses for the crosshead pin bearing, and having one or two guide plates connected to it, according to whether the engine has single or double guides. Such a crosshead for a single slipper guide is seen in Fig. 187, the guide plate or

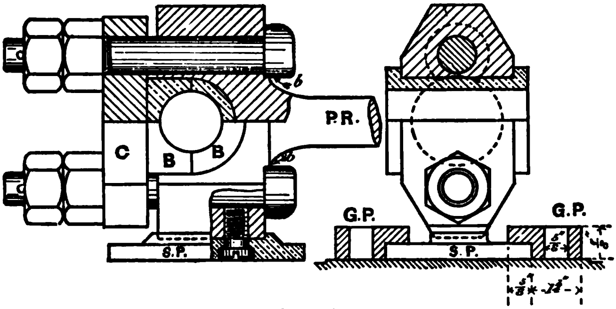


Fig. 187.

"*slipper*" of cast iron or gunmetal being fixed by set screws to the under side as shown; the brasses B B are held in position by two bolts, *b b*, and a steel cap, C, the crosshead and cap being cut away at an angle of 60° around the bolts to save weight. In this example, which is for the vertical engine of Ex. A, the crosshead and the piston-rod, P R, are forged in one piece of mild steel. For such a crosshead the small end of the connecting-rod is simply forked like an ordinary pin or knuckle joint as shown in Fig. 178 (see § 59), having the crosshead pin fixed to it. The slipper plate, S P, moves between guides formed by bolting two plates, G P, to a raised part of the engine framing, the cross-section being as shown in the end elevation of Fig. 187 (see also Fig. 178, D). This arrangement is often spoken of as a single slipper guide.

In the above design it will be noticed that the brasses are part of the crosshead, and the crosshead pin part of the connecting-rod, but in other patterns now to be described the brasses are a part of the connecting-rod, and the crosshead pin of the crosshead.

In the design of Fig. 188 the crosshead is a cast-iron box, C B, having the top and bottom faces circular and turned to an arc, having its centre at the centre line of the piston-rod, P R. The

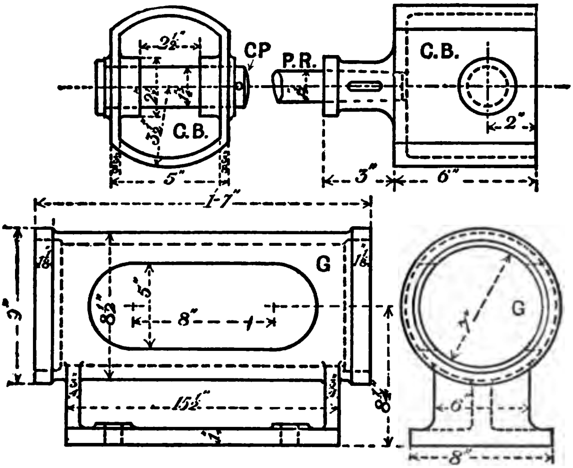


Fig. 188.

piston-rod is connected by a cottered joint, and the crosshead pin, C P, passes through bosses cast on the inside of the crosshead, the distance between them being equal to the length of the brasses in the small end of the connecting-rod. The guide G is then a cast-iron trunk having part of the sides removed, and bored inside to receive the crosshead, and with flanges attached

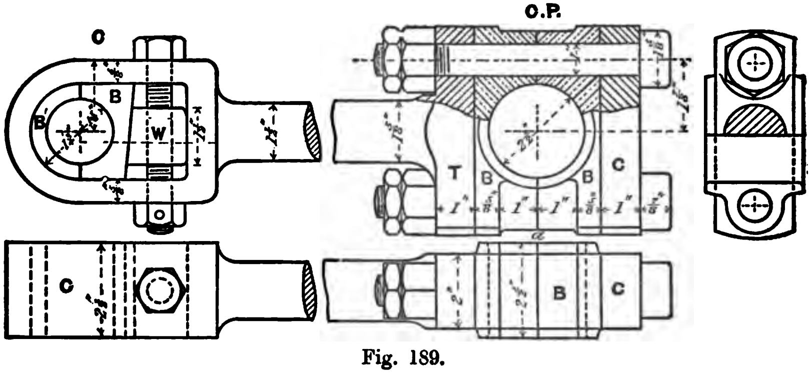


Fig. 189.

for bolting to the engine bed. The diameter of the guide inside, and, therefore, of the crosshead, is so arranged that the connecting-rod shall clear the guide when in its highest and lowest positions. The crosshead and guides as dimensioned in the figure are for a horizontal engine, cylinder 8″ diameter, and 10″ stroke.

The chief advantage of this form of trunk guides is that the machining can all be done in the lathe or boring machine, and is thus much cheaper than the planing and shaping required for the slipper guide, and for the guides of Fig. 190. This type of trunk guide is, therefore, very common in ordinary commercial engines, both vertical and horizontal.

The box form of crosshead with its inside pin necessitates a special arrangement of the small end of the connecting-rod, as

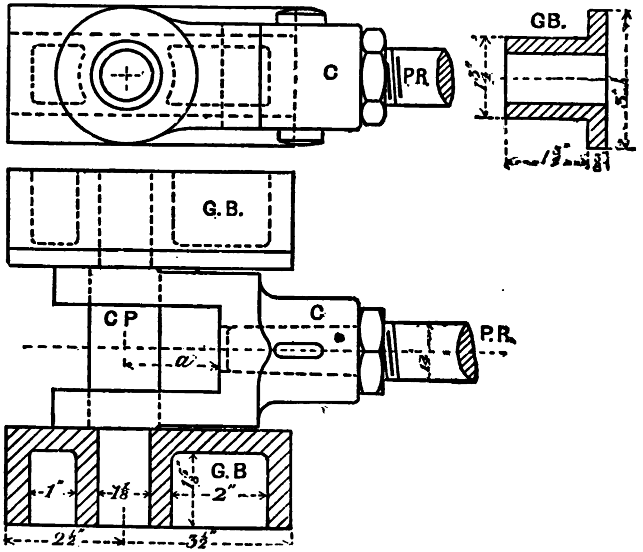


Fig. 190*a*.

it is evident the brasses could not be adjusted from inside the pin. The design to meet this difficulty is shown in Fig. 189, end C; the rod is forged with an enlarged end, which is then slotted out to receive the brasses, B B′, and the wedge block, W, which is moved up and down against the tapered face of the front brass, B, by turning round the screw shown, thus adjusting the brasses. The back brass, B′, is a semicircular bush often provided with a pin to prevent movement, but this is really

unnecessary, as there is no tendency for the brass to move sideways when in position.

In the third design, shown in Fig. 190*a*, the crosshead C is a wrought-iron fork or knuckle joint, fastened to the piston-rod P R by a cotter, and holding the crosshead pin C P, thus making an ordinary pin joint (see § 59). The crosshead pin is extended on each side, and fits into cast-iron guide blocks, G B, which move between double guide bars, G (Fig. 190*b*) of **T** section, bolted at one end to a projection on the cylinder cover (Fig. 181), and at the other to a vertical frame, G F (Fig. 190*b*). This is also a common arrangement for horizontal engines, but is more expensive, although no more convenient than the slipper guides of Fig. 187. It possesses one advantage in common with the slipper guide as compared with the trunk guides, in allowing freer access to the piston-rod gland and to the crosshead brasses.

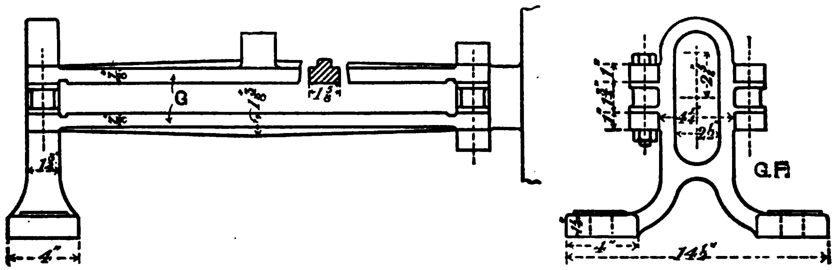


FIG. 190*b*.

The design of the small end of the connecting-rod for this type of crosshead will be explained under the heading of connecting-rods. The crosshead and guides of Fig. 190*a*, *b* are for the horizontal engine of Ex. B.

(181) **Connecting-Rod Ends.**—The ends of connecting-rods which join the crank pin are provided with brasses surrounding the pin, held in position in two different ways. In one design shown in Fig. 189, the connecting-rod end, C P, is enlarged, making what is called a **T** end (marked T), and the brasses B B are fixed between it by bolts passing through them and through a cap C, the nuts being on the rod end, as in the figure, or, as frequently seen, on the outside of the cap. The brasses are adjusted for wear by filing away the common face of contact at *a*, or by thinning a packing piece which is often fitted between the brasses. In a similar and very economical design, the rod is forged with a large solid end, which is first bored for the brasses and bolts, and then slotted across the centre to

separate a piece for the cap; the brasses are then simple half bushes or rings, held in position by being just cut away at the bolt holes, and, therefore, being held by the bolts. In the arrangement of Fig. 191, the rod end is enlarged to an oblong section, and the brasses B and B′ are held between the end and the inside of a strap, S, which surrounds them and the oblong portion of the rod end, the strap is fixed by a cotter, C, and a gib, G, so that when the cotter is driven in, the gib draws the

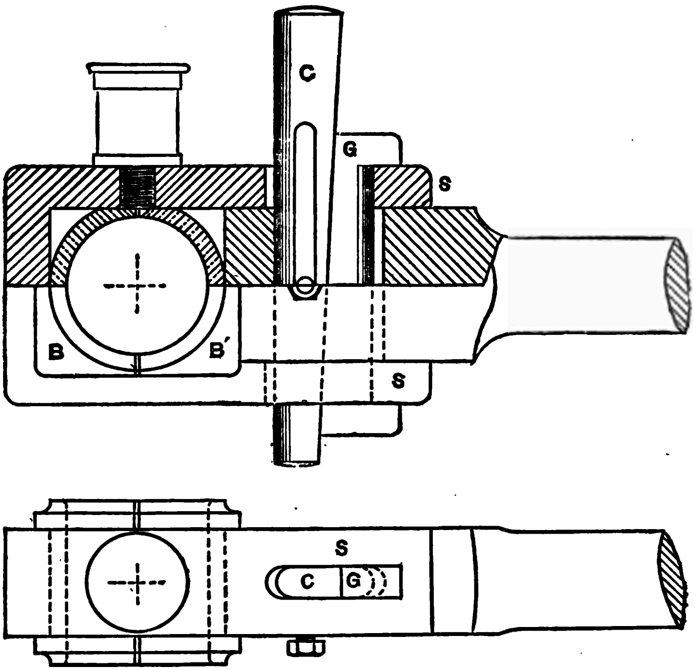


Fig. 191.

strap towards the rod, and brings the brass B nearer the brass B′. This use of a gib and cotter has already been referred to in § 67, but owing to the cost of machining, it is more expensive than the design with bolts and nuts. It has recently been cheapened by cutting the slot for the cotter quite at the rod end, so that there is no metal between the brass and the slot. The strap is then rigidly bolted to the rod through a part removed from the brasses, and the gib is placed next the brass B′, and

bearing against it, so that when the cotter is driven in it bears hard against the rod and strap on one edge, and on the other against the gib moving it and the brass B′ towards the brass B.

When a rod has its crank pin end of the design of Fig. 191, its crosshead end is generally similar, and a rod so made is used in the horizontal engine of Ex. B, the crank pin end of the rod in Ex. A being of the form of Fig. 189, end C.P. (see Fig. 178).

(182) **Lubrication of Connecting-Rod Brasses.**—In Fig. 191 a siphon oil cup (see Fig. 153) is screwed into the top of the strap over the centre of the brasses, and oil is conducted to the front and back parts of the brasses which we know to be the seats of pressure, through narrow grooves cut in the top halves, leading from the hole under the oil cup; a horizontal groove being also cut nearly across the brasses exactly at the centre line. For vertical engines (say for the crank pin end) it is customary to fix a separate oil cup to the connecting-rod, and lead a small pipe from it to a hole drilled through the T end of the rod and the brass next it, a groove being cut from this hole around the brasses to the other side of the journal.

(183) **Proportions of Parts.**—In obtaining the sizes for the crossheads, guides, and connecting-rods, we require to know the total forces acting upon them when working. The stresses due to the steam pressure upon the piston alone are easily found, but these are increased by the effect of the inertia of the parts,

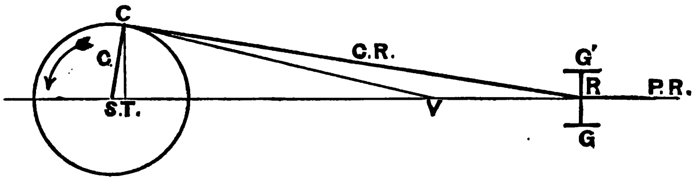


Fig. 192.

especially with high speeds, to an extent which must not be neglected, but which can only be found for particular examples by a method far too complicated for insertion here. We can, however, usefully show how to find the stresses caused by the steam pressure, and then by allowing a low value for the working stress per square inch, give some idea of how the sizes of the parts are obtained.

In Fig. 192, P R is the piston-rod, G and G′ the guide blocks, C R the connecting-rod, and C the crank, S being the shaft

centre. (The crank and connecting-rod are drawn in the position when at right angles to each other.) If C T be drawn at right angles to R S, and, therefore, parallel to the pressure on the guides, then if the length of R T represents the total pressure on the piston, the length of C R will represent the force along the connecting-rod, and of C T the pressure on the botton guide. By drawing arrows representing the direction of the forces, it will be seen that in both the inward- and outward-stroke the guide pressure is on the bottom guide G when the engine runs in the direction of the arrow, the rod being subject to tensile stresses during the inward-stroke, and to compressive stresses during the outward-stroke. We thus see that a top guide is unnecessary so far as the pressure upon it is concerned, except when the engine reverses Now draw C V to represent a shorter connecting-rod, then for the same position of the crank we see that, if the length of V T represents the total pressure on the piston, the length of C T represents, as before, the pressure on the guide. But this means that the guide pressure is the same as before, with a less pressure on the piston, or what amounts to the same thing, the *shorter* the connecting-rod, the *greater* the pressure on the guides. A short connecting-rod also gives a more irregular twisting force at the crank pin than a long rod, so that as long a connecting-rod as is conveniently possible should always be secured.

(184) **Length of Connecting-Rods.—The length of connecting-rods between the centres for such engines as we are dealing with varies from five to seven times the crank length, the usual standard being six times.**

Turning again to Fig. 192 we see that since the angles R C S and R T C are right angles, the triangles S C R and C T R are similar (C R being common), that is—

$$SC : CR : RS :: CT : TR : RC,$$

also that

$$RS = \sqrt{RC^2 + CS^2}, \text{ which } = \sqrt{\text{con.-rod length}^2 + \text{crank length}^2},$$

and if the rod is 6 cranks length this equals $\sqrt{37}$, or 6·08. So that as R C is the total pressure on the piston when R S is the force along the connecting-rod, we see that with the usual length of rod the maximum force along the rod may be taken as equal to the total pressure on the piston (or more exactly to $\frac{6{\cdot}08}{6} = 1{\cdot}013$ times the piston pressure); also as C S represents the maximum guide pressure we have, that this is equal to $\frac{CS}{CR}$

or $\frac{1}{8}$th of the total pressure on the piston, or in a general case is equal to $\frac{\text{length of crank}}{\text{length of rod}} \times$ total pressure on piston.*

As the piston will not have reached the point of cut-off in the engines we are dealing with, when the rod and crank are at right angles, the pressure on the piston will be taken at its greatest value—that is, $p \times d^2 \times \frac{\pi}{4}$, where p = effective pressure of steam per square inch before cut-off, and d = piston diameter.

We see then that the force along the connecting-rod may be taken as equal to the maximum total effective pressure on the piston when the steam alone is considered, and that this force has to be resisted by the crank pin and crosshead brasses. But, as previously mentioned, we must allow for the stresses due to inertia, and this we shall do by taking such a value for the working stress as shall cover these stresses, including those due to the steam. Professor Unwin has shown that the total stress in the connecting-rod is from $1\frac{1}{4}$ to $1\frac{1}{2}$ times the greatest pressure on the piston, as caused by the steam, but that the maximum guide pressure is not much increased.

(185) **Area of Rubbing Surfaces.**—The following figures for the working pressure upon the rubbing surfaces are ordinary practical values, and are in *lbs. per square inch.* The total pressure upon the surface is to be taken as that caused by the maximum effective steam pressure upon the piston only. The required area in square inches will, therefore, in all cases be equal to $\frac{d^2 \times \frac{\pi}{4} \times p}{s}$, where d = piston diameter in inches, p = maximum effective steam pressure per square inch, s = working pressure per square inch upon the guide.

(186) **Guide Blocks.—The maximum pressure should not exceed 50 lbs.** In many cases it is much lower, as other considerations in the design may require a large surface, such, for example, as in the box crosshead and trunk guide of Fig. 188.

(187) **Crosshead Pin.**—The effective area is only that of the length in the brasses, and is, therefore, that of the brasses also,

* If the angle C R T = ϕ, and T R = total effective pressure on piston = P, then—

$$\text{R C} = \text{force along connecting-rod} = \frac{\text{P}}{\cos\phi}$$
$$\text{C T} = \text{pressure on guide} = \text{P} \,.\, \tan\phi,$$

and these have their maximum value assuming P to be constant, when the angle R S C is a right angle.

which as already seen may be either in the crosshead or the connecting-rod. **The maximum pressure should not exceed 2,000 lbs., a good working value being 1,200 to 1,500 lbs.** If d = diameter of the pin, and l = length embraced by the brasses in inches, then effective area = $d \times l$ square inches.

(188) **Crank Pin.—The maximum pressure should not exceed 600 lbs., a good working value being 400 lbs.** If d = diameter and l = length in inches, then effective area = $d \times l$ square inches.

(189) **Ratio of Diameter and Length of Crosshead and Crank Pin.**—The calculation of the diameter and length of a crosshead or crank pin in order that the pressure upon them shall not produce too great a stress in the material, is another one of the questions which we cannot enter into here. We may, however, indicate that an ordinary crosshead or crank pin is practically in the condition of a beam loaded uniformly and supported at each end (assuming the crank to have double webs), and is, therefore, subject to bending, the bending moment increasing with the length, a crank pin being also subject to a twisting action. These considerations are fully treated in Prof. Unwin's "Machine Design."

Crosshead pins usually have a length of from $1\frac{1}{4}$ to $2\frac{1}{4}$ times their diameter, an average ratio being 2, or $l = 2\,d$.

Crank pins usually have a length of from 1 to $1\frac{1}{8}$ times their diameter, an average ratio being $1\frac{1}{8}$, or $l = 1\frac{1}{8}\,d$.

EXAMPLES.

EX. A 4. Make working drawings, two views, full size, of the crosshead and slipper guide for the vertical engine of Ex. A, as in Fig. 187. The crosshead block to be of mild steel forged in one piece, with the piston-rod $1\frac{1}{4}''$ diameter. Length of block, $2\frac{3}{4}''$. Pressure on crosshead pin 674 lbs. per square inch, $l = 2\frac{1}{4}\,d$, brasses $\frac{3}{8}''$ thick. Two bolts, total area at bottom of threads = area of piston-rod. Cap $1\frac{1}{8}''$ thick. Guide plate $\frac{7}{16}''$ thick, $3\frac{3}{4}''$ wide, pressure per square inch 35 lbs., fixed to crosshead by 2 $\frac{5}{8}''$ screws. Lock nuts on bolts. Height from centre of rod to base, $3\frac{5}{8}''$.*

(Calculate length and diameter of brasses; these have flanges $\frac{7}{16}''$ thick overlapping crosshead, therefore width of crosshead = $l - \frac{7}{8}''$. Draw in brasses in both views. Find diameter of bolts taking effective area as at bottom of threads. Bolts are kept from moving by turning the crosshead where the rod joins it, forming a collar of about $2\frac{1}{8}''$ diameter, and then

* The following are the pressures per square inch to be worked to for the examples of A and B:—Ex. A—Initial pressure, 107 lbs.; back pressure, 17 lbs.; effective initial pressure, 90 lbs. Ex. B—Initial pressure, 98 lbs.; back pressure, 2 lbs.; effective initial pressure, 96 lbs.

cutting away the bolt heads as shown at b, to fit against the collar, this means that the bolts are within about $\frac{1}{8}''$ of the brasses; thus draw in centre of bolts to nearest $\frac{1}{8}''$. Make length of block from head of bolts to centre $1\frac{7}{8}''$, and from centre to front face $\frac{7}{8}''$. Draw in cap, allowing $\frac{1}{4}''$ clearance. Amount of metal outside bolt to be enough for surface of nut on cap, and about $\frac{3}{4}''$ on block. Thickness of slipper plate where bolted to cap $\frac{9}{16}''$. Slipper thickened to $\frac{3}{4}''$ at centre and recessed $\frac{1}{8}''$ to receive crosshead block. Cut away top and bottom of block at 60°, and show the rod meeting the block with a gradual curve. Set screws for slipper plate should not touch the lower bolt, and heads must be recessed within the plate.)

EX. B 4.—Make working drawings, full size, of the crosshead guide blocks and guides (two or three views of each) for the horizontal engine of Ex. B, as in Fig. 190. Forked crosshead cottered to $1\frac{1}{4}''$ piston-rod. Pressure on crosshead pin, 1,938 lbs. per square inch, $l = 1\frac{1}{4}\,d$. Guide blocks, $1\frac{3}{4}''$ wide; pressure, 43 lbs. per square inch; $1\frac{3}{4}''$ between guide bars; inside flanges on blocks, $\frac{3}{8}''$ thick; crosshead pin through guides, $1\frac{1}{8}''$ diameter. Double guide bars of cast iron, total length 2′ 4″, bolted at back end to cylinder and at front end to standard, as shown, with $\frac{3}{4}''$ bolts.

(Calculate sizes of crosshead pin as before. Then design forked and cottered joint as in §§ 65, 66, making the distance $a = 2\frac{1}{2}''$ to allow for connecting-rod end; stress on rod at weakest section, 7,000 lbs. per square inch, due to piston pressure. Find length of guide blocks for given pressure and width. Draw crosshead pin out of centre of guide block, as shown, to allow a greater clearance between gland and crosshead. Metal of guide blocks, $\frac{1}{4}''$ thick. The inside top and bottom flanges on the guides prevent side play. Length of guide bars to allow for stroke, block, $\frac{1}{4}''$ clearance each end, and for bolting to cylinder and front frame. The bars are $\frac{7}{8}''$ thick, strengthened by a rib, $\frac{3}{4}''$ wide and $\frac{1}{2}''$ thick at the centre. In calculating size of guide bars for the thrust due to greatest piston pressure, they should be treated as beams supported at each end. The bending force is the maximum thrust on the guide, the position of the guide centre, relative to the ends of the bars when the crank and connecting-rod are at right angles, can be found by construction, and gives the distance of the bending force from the beam ends, and then the size can be found, knowing that B M $= f\,Z$, where $f = 3{,}000$ lbs. per square inch, and Z for rectangular section $= \frac{b\,d^2}{6}$, the effect of the strengthening web, when small, being neglected. The upper bars might be made much smaller, but for convenience in casting all alike. The sizes of the front standard are marked in Fig. 190b, the inside width must allow clearance for the connecting-rod at the sides, and the inside height at top and bottom, when the rod is in highest and lowest position. Height of centre line from base to be obtained from cylinder drawing, Fig. 181, also details of lugs on cylinder cover to which guide bars are bolted.)

Connecting-Rods (EX. A 5).—Make working drawings, full size, of the connecting-rod for vertical engine of Ex. A. Crank pin end with solid brasses and cap, as in Fig. 189, end C P, crosshead end forked, as shown in Fig. 178. Length of rod, 6

cranks; diameter at crank pin end, $1\frac{9}{16}''$, tapering to $1\frac{5}{16}''$ at fork; pressure on crank pin, 352 lbs. per square inch; $l = 1{\cdot}16\, d$. Size of crosshead pin from Ex. A 4. T end of connecting-rod and cap for crank end, $2\frac{7}{8}''$ wide, $1\frac{1}{8}''$ thick. Sectional area of bolts at bottom of threads = area of rod at smallest section. Thickness of brass next cap and rod, $\frac{5}{8}''$. Smallest section of fork, $1\frac{1}{2}''$ wide, $\frac{5}{8}''$ thick; thickness where pin passes through, $1\frac{1}{4}''$. Show oil cups as in the figure.

The large end of rod should be drawn half in section. It is unnecessary to show the whole length of the rod, so that rod can be shown broken off and the ends drawn near together. The calculations for the rod diameter are too difficult for explanation. First, find the size of crank pin, thus obtaining length of brass, then draw in lines of the T end of the rod, the brasses and cap in front elevation. Find diameter of bolts, and draw them in, they may be within $\frac{1}{8}''$ of journal. Let the width of the brasses be equal to the width of rod and cap for a length of about half the cap thickness, then remove metal around the bolts, as shown. Bolts kept from turning by cutting away part of the round heads, and of cap as for bolts in Fig. 193*a*. Length of cap and T end of rod to allow surface for nuts, as in ordinary flanged joint. Show a gradual curve to the T end. The fork at the crosshead end must be long enough to clear the crosshead, its other proportions will present no difficulty. The crosshead pin is kept in position by a keep pin through one fork. See pp. 135, 136 for the curves at the T end of the rod and at the fork.

EX. B 5.—Make full-size working drawings of the connecting-rods for horizontal engine of Ex. B. Both ends to have straps with gibs and cotters, as in Fig. 191. Length of rod, 6 cranks; diameter, $2\frac{1}{4}''$ from crank end for half length, then tapered to $1\frac{3}{8}''$ at crosshead end; pressure on crank pin, 426 lbs. per square inch; $l = 1{\cdot}2\, d$. Size of crosshead pin from Ex. B 4. Brasses to overlap at crank end $\frac{11}{16}''$, at small end $\frac{1}{4}''$ on each side; thickness at seat of pressure, $\frac{7}{16}''$ and $\frac{5}{16}''$ respectively. Mean width of gib and cotter combined at crank end $2''$, at small end $1\frac{1}{2}''$; cotter, $1\frac{7}{8}''$, and $1''$ within rods at each end respectively, other proportions as in § 67. Thickness of strap at crank end $1''$, at small end $\frac{11}{16}''$.

(Calculate size of pin. Draw in circle showing diameter of pin, and then show thickness of brasses; as the pressure on the rod acts in the direction of its length, the brasses only wear at front and back, and not at top and bottom near the strap, hence the brass there may be thinner; this gives distance between strap. Show thickness of strap, then obtain sizes of cotter and gib (§ 67), and draw in position, allowing sufficient length of rod and strap beyond the cotter way. The brasses are to have flanges at each side to prevent side play, but these need not exceed $\frac{3}{8}''$ thick. Show arrangement for lubrication.

We have seen that the size of the brasses decides the distance between the strap, and also to some extent its width, since too much of the brass must not overhang. It would then seem as if the strap thickness and the sizes of gib and cotter should be proportioned to resist the stresses due to

the steam pressure and the other forces acting along the rod. But if this is done, the strap, gib, and cotter would be much smaller than actually made, even taking as low a stress as 3,000 lbs. per square inch. It, therefore, seems as though the parts have been proportioned more for symmetry than for equal strength, but there seems to be no reason why the student should not work out the sizes for himself, remembering that the strap must be thick enough, so that the bearing pressure upon it may not be too great, and adopting for convenience the usual rule that the cotter thickness $= 0{\cdot}3\,d$ where d = diameter or width of rod through which cotter passes, and where f = 5,000 lbs. per square inch for wrought iron.)

SECTION XXIX.

ECCENTRICS AND ECCENTRIC-RODS.

(190) **Eccentrics.**—We shall now consider the form and design of engine eccentrics, referring only to their application in working the slide valves. An eccentric consists of two parts, the "*sheave*," which is a disc, usually of cast iron, rigidly keyed to the crank shaft and turning with it, and the "*strap*," a cast-iron, wrought-iron, or gunmetal ring which surrounds the sheave, and is attached to one end of the eccentric-rod, the other end of which connects with the valve-rod. When the sheave can be

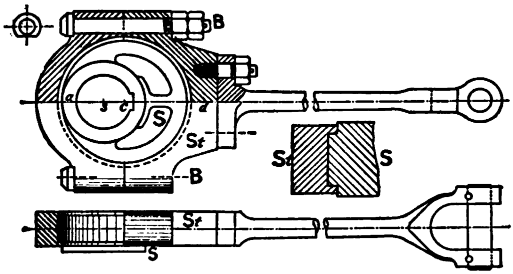


Fig. 193*a*.

slipped upon the shaft from one end, it is cast in one piece, but when it cannot be so moved, it is cast in two parts, which are bolted together in position. The strap is in two parts, connected by bolts, one half being fixed to the eccentric-rod, generally by means of studs and a T end. The eccentric-rod joint, with the valve-rod, is almost always a simple pin and fork connection, and often without brasses, as the movement is so very slight.

(191) **Forms of Eccentrics.**—Examples of eccentrics and rods are shown in Fig. 193*a*, *b*. The shaft centres are marked

s, the sheave centres c, the sheaves S, and the straps St. The rods are made with T ends fixed to projections on the sheave by two studs, as shown in the figures; and with a short turned piece, which fits a hole in the eccentric strap, thus preventing any side movement, and relieving the studs of any tendency to shear. In Fig. 193*a* the sheave is cored out as shown to save weight, and in both cases the sheaves are widened next the shaft, to give a greater length for the keys. The two halves of the strap are connected by bolts, B B, which may be placed as close to the sheaves as $\frac{1}{8}''$ or $\frac{1}{4}''$. It will be seen that the straps of Fig. 193*a* have much thicker bolt flanges than those of Fig. 193*b*, and that the bolts can, therefore, be brought close in, such a design is also more rigid, and is, therefore, preferable.

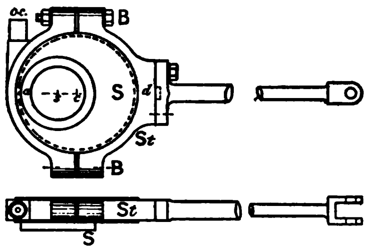


Fig. 193*b*.

The straps are kept in position on the sheaves by small side flanges, which bear against the sides of the sheave, as shown in the separate figure of Fig. 193*a*, where S is a section of the sheave, and St of the strap. Care must be taken to provide for an efficient lubrication of the rubbing surfaces, either by casting an oil cup on one half of the sheave, as at O C in Fig. 193*b*, or by other means. In each case the eccentric-rod ends where joined to the valve-rods, have the form of forked pin joints. Eccentric-rods are often made of cast iron, and in one piece with one half of the sheave, the rods having a double tee section. Wrought-iron and steel rods are also made of rectangular section.

(192) **Forces on Eccentrics and Rods.**—The chief forces acting upon the eccentric-rod, and also upon the sheave and strap, are caused by the resistance offered by the slide valve to movement, and have already been considered in § 174. As in the case of valve-rods, a certain size of eccentric is necessary from quite other considerations than those of strength, and it is probable that with small engines, the proportions necessarily adopted give ample sizes for the forces which come upon them, but it is at all times advisable to see what these actually amount to.

(193) **Pressure on Sheave.**—If d = diameter of sheave and w = width embraced by the strap, then $d \times w$ = effective bearing area resisting the force along the rod. In practice the pressure per square inch upon the sheave due to this force is about 90 to 100 lbs., thence $F = d \times w \times$ pressure per square inch.

(194) **Size of Sheave.**—The eccentricity or throw of the sheave must be equal to half the travel of the valve (see § 173). Therefore in Fig. 193*a*, *b*, if *s* is the shaft centre, and *c* the centre of the sheave, then the length between *s* and *c* must equal the eccentricity or throw; this length is also sometimes called the "radius of the eccentric." The movement is the same as though a crank of equal length were used, hence the stroke or travel of the valve is equal to twice the throw of the sheave. The thickness of metal, *a*, from the edge of the hole to the outside of the sheave should be as small as possible consistent with strength, in order to obtain a small diameter of sheave, and may be equal to $\frac{1}{4}d$ where d = shaft diameter, the minimum thickness being $\frac{5}{8}''$. Then the diameter of the sheave is easily found, and its width determined in order to give sufficient bearing pressure, but remembering also that it must have a greater width than the diameter or breadth of the eccentric-rod, as will be seen in designing the strap. Eccentric sheaves usually have a width next the strap of $\frac{1}{5}$th to $\frac{1}{4}$th their diameter.

(195) **Position of Sheaf upon Crank Shaft.**—The position of the eccentric sheave upon the crank shaft relatively to the crank is found as follows:—In Fig. 194, let C be the shaft centre and C A the position of the crank at the back dead centre; draw a circle, A E B, having a radius, A C, equal to the throw or eccentricity of the crank. Along A B from C set off C L = steam lap of valve, and L M = lead of valve. Draw a line through M perpendicular to A B to meet the circle in the point E; then when the crank is moving in the direction of the arrow, the line C E is the centre line of the sheave, and E is the centre of the sheave. The point *f* would be the sheave centre without lap or lead, and the point *g* with lap but without lead.

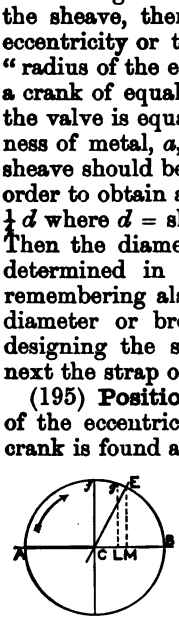


Fig. 194.

(196) **Size of Strap.**—The width of the strap is usually made $\frac{1}{8}''$ to $\frac{1}{4}''$ less than the width of the sheave to allow for clearance, especially when two eccentrics are next one another. Wrought-iron or steel straps have a thickness of $\frac{1}{2}w$, gunmetal straps of $\frac{5}{8}w$, and cast-iron straps of $\frac{3}{4}w$, where w = width. The least thickness of the flanges should be $\frac{1}{4}''$, the depth being usually about $\frac{1}{16}''$ greater. The length of the part marked *d* (Fig. 193) to which the rod is attached must be enough to give a sufficient depth for the studs, and its length in the other direction to give room for the nuts on the T end of the eccentric-rods.

(197) **Size of Studs.**—These may be calculated knowing that the tensional stress at the bottom of the thread is equal to the

total stress in the rod, or the effective combined area of the studs should be equal to area of rod at smallest section. But their diameter should not exceed $\frac{1}{2}$ w, otherwise there will not be sufficient metal around the holes.

(198) **Size of Eccentric-Rods and Valve-Rods.**—We are not able to show how to find the diameters of these rods, for reasons already given. Like connecting-rods, eccentric-rods are tapered from the eccentric to the valve end, and must be designed as struts. The T ends of the rods are equal in width to the strap, and in length to allow room for the nuts, the thickness being generally from d to $1\frac{1}{4}$ d, where d = diameter of studs. The pin and fork connection at the valve-rod end should be proportioned according to the method of § 59, taking the total force upon it as equal to the frictional resistance of the valve (§ 174).

(199) **Valve-Rod Guides.**—The guides for valve-rods are usually simple cast-iron brackets, bolted to the steam chest or engine bed, and having bosses through which the rods work, the rods being enlarged in diameter for a length equal to travel of valve + length of guide. Owing to the small amount of movement it is generally unnecessary to bush the guides, or to arrange for taking up wear, but provision must be made for lubrication. In the vertical engine of Ex. A (Fig. 178), the guide bracket is bolted to the under side of the steam chest, and in the horizontal engine of Ex. B (Fig. 179), it is bolted to the bed-plate in the position shown. The form and position of the bracket should be left over until a general arrangement drawing is made, and the student should then be able to design it without any difficulty.

EXAMPLES.

Make working dimensioned drawings, showing two views, elevation and plan, part of each in section, full size, of the following eccentrics and rods, and valve-rod ends:—

EX. A 6. Eccentric and rod as in Fig. 193*a* for the vertical engine of Ex. A. Diameter of shaft $2\frac{3}{8}''$, throw $\frac{7}{8}''$, diameter of sheave $5\frac{3}{4}''$, width $1\frac{1}{2}''$ recessed to $1''$ as shown, increased on one side only to $1\frac{3}{4}''$ next shaft. Strap $\frac{3}{4}''$ thick next bolts to $\frac{7}{8}''$ thick at centre, flanges for sheave $\frac{3}{16}''$ thick, $\frac{3}{16}''$ deep, sheave and strap of cast iron. Eccentric-rod $\frac{3}{4}''$ diameter at valve-rod end to $\frac{7}{8}''$ diameter at eccentric end. Valve-rod end as in Fig. 178, diameter of pin $1''$, length between forks of eccentric-rod $2''$, end of valve-rod to have brass bush $\frac{1}{4}''$ thick, enlarged part of rod for guide $3\frac{1}{2}''$ long $1\frac{1}{8}''$ diameter, nearest end to eccentric-rod pin centre, $1\frac{3}{4}''$ away, length of part of rod ($\frac{7}{8}''$ diameter) from end of enlarged

part to under side of piston valve $6\frac{7}{8}''$, thickness of arms of fork surrounding eccentric-rod pin $\frac{3}{4}''$, pin held by keep pin. Show key for fixing sheave to shaft (see § 71).

(Draw in shaft, sheave, and strap, then decide bolts for fixing strap, and for fixing rod to strap, of the same diameter, and so that area at bottom of thread of 2 bolts shall equal area of rod at smallest part; bolts for sheave may come within $\frac{1}{8}''$ of strap, then arrange length of strap around bolts, so that sufficient room is obtained for the nuts without cutting away too much of strap. Bolts to have round ends cut away to prevent turning round, as shown. Lock nuts and keep pins. Arrange sufficient metal beyond sheave next rod so that studs holding rod to sheave may be screwed in a depth of $1\frac{1}{4}\,d$; holes for studs not nearer to sheave than $\frac{3}{8}''$. Next design T end of rod, the junction to the rod being by a gradual curve, and afterwards complete end of rod next valve-rod and valve-rod. Exact length of rods left till general arrangement drawing is made. See pp. 135, 136 for curves of T end of rod and of forked end.)

EX. B 6. Eccentric and rod as in Fig. 193*b*, and slide rods as in Fig. 185, for the horizontal engine of Ex. B. The eccentric and rod for the main valve only need be drawn, the expansion valve eccentric, which differs only in having a longer throw, being left till making the general arrangement drawing. Diameter of shaft $3\frac{1}{2}''$, throw $1\frac{1}{8}''$, diameter of sheave $7\frac{7}{8}''$, width $1\frac{1}{2}''$, increased on one side only to $2''$ next shaft. Strap $\frac{3}{4}''$ thick, flanges for sheave $\frac{3}{16}'' \times \frac{3}{16}''$. Sheave of cast iron, strap of gunmetal. Eccentric-rod $\frac{3}{4}''$ diameter at valve-rod end to $\frac{7}{8}''$ at eccentric end. Valve-rod end as in Fig. 185, with ordinary fork and pin joint, pin $\frac{5}{8}''$ diameter, $\frac{7}{8}''$ between jaws of fork. Enlarged part of rod for guide $1\frac{5}{8}''$ diameter, $5\frac{5}{8}''$ long, rod at other parts $\frac{5}{8}''$ diameter. Show key for fixing sheaves to shaft, and oil cup on eccentric strap.

(Proceed as in the last example, for drawing the bolts through the strap (ordinary hexagonal heads, with washers on each side), and for the T end of the eccentric-rod. Then design the forked end for the valve-rod. Details of valve-rods are shown in Fig. 185, and both rods should be drawn. The main valve-rod is screwed into one end of a larger rod, G R, which forms the guide, the opposite end being shaped to a single eye, for fitting between the fork of the eccentric-rod. The guide, E G R, for the expansion valve is of cast iron, having one end shaped as shown so that the valve-rod can pass through the guide and be fixed by nuts, thus allowing a better means of adjusting the position of the expansion valves; the extreme end is then made into a single eye for connecting to the eccentric-rod. In the figure the guides, G R and E G R, are drawn as though the centre lines of the two rods were in a vertical plane, one above the other, whereas since the slide valves are on a side of the cylinder they are really in a horizontal plane, one within the other. Hence the guides should each be turned round through 90°, and when this is done the two guides clear each other at all parts of their movement. The length of the rods can be left until making the general arrangement drawing.)

SECTION XXX.

CRANK SHAFTS AND BEARINGS.

(200) **Crank Shafts.**—The cheapest form of engine crank shaft is made by keying a cast- or wrought-iron disc to a length of shafting, and fixing a turned steel or wrought-iron pin (called the crank pin) in the disc at a distance from the shaft centre, equal to half the engine-stroke. This arrangement is termed an overhung disc crank pin, its advantage being that it is cheap to construct, requires only one bearing, and, therefore, reduces the size of the engine bed plate, and helps to balance the connecting-rod end, especially when the disc is cored out and lightened

Fig. 195.

around the pin. The ordinary single-cylinder horizontal engines, with trunk guides, frequently have this form of crank. In another form of single-arm overhung crank, the crank arm or web is of rectangular section, first shrunk on, then keyed to the shaft, the pin being a separate piece to the crank web. Such

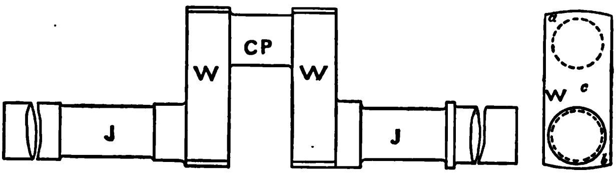


Fig. 196.

cranks are termed "*built up.*" Solid cranks have two webs, the pin and webs being in one forging with the shaft. The cheapest make of solid crank is that shown in Fig. 195, where a shaft of uniform diameter is bent to form the webs and pins, and afterwards turned at the parts for the journals. Small power engines are frequently fitted with cranks of this make.

The form of solid crank which is most commonly used is shown in Fig. 196. The webs, W W, are of rectangular section, the corners where the crank pin, C P, and shaft join being well rounded. The top and bottom of the webs are turned by fixing the shaft in the lathe so that the centre line passes through the centre of the length of the webs, and, therefore, in the end view, *c* is the centre for the arcs *a* and *b*. Journals for the bearings are formed by turning the shaft at J J to a smaller diameter for the required length, thus providing collars which prevent side movement. The webs are often lightened by removing the corners at an angle of 45°.

(201) **Counterbalance Weights.**—It is very desirable, especially in high-speed engines, to balance the crank pin and webs, and a part at least of the connecting-rod end. This is effected by blocks of metal, called counterbalance weights, attached to the crank webs at the ends opposite the crank pin, as seen in the engines of Ex. A and B (Figs. 178 and 179). Counterbalance weights may be solid, with the shaft as in the figures, or may be fixed to the crank webs by straps. The design of counterbalance weights will be dealt with in Section xxxiii.

(202) **Size of Crank Shafts.**—We are not able to enter into the conditions which decide the proportions of crank shafts, owing to their complex character. Crank pins and webs are subject to direct tension, and also to bending and torsional stresses, caused both by the pressure on the piston and the inertia of the moving parts. The shaft diameter at its smallest part can be found by the ordinary rule for shafts. $\text{T M} = 0{\cdot}196\, d^3 f$, where T M = twisting moment, d = shaft diameter, and f = working stress per square inch, which may be taken as 5,000 lbs. for mild steel, in order to prevent too great an angle of twist in the shaft length. But the twisting moment must be the maximum during the stroke, and can only be correctly found by allowing for the inertia of the moving parts.

(203) **Crank-Shaft Bearings.**—The drawing of a crank shaft cannot be completed until the size of its journals has been decided. The diameter of the journals is settled by knowing the diameter of the shaft, so that we have only to find the length, and we know this must be sufficient, so that the pressure upon the brasses per square inch shall not exceed a certain value. The whole question of the design of lubricated bearings has been fully discussed in Section xi., and need not, therefore, be repeated here. For engines such as we are dealing with, practical experience allows a pressure per square inch upon the crank-shaft journals of from 100 to 200 lbs., due to the pressure on the piston only, an average value being 150 lbs. When a flywheel is fixed

on one end of the shaft, the bearing next to it should have an increased length.

(204) **Forms of Crank-Shaft Bearings.**—Crank-shaft bearings are often cast as a part of the engine framing, this being the common construction in horizontal engines with overhung crank pins, and with vertical engines. In other cases the bearing is a separate pedestal bolted to the bed plate. In the vertical engine of Ex. A, the bearings are formed in the bottom box framing, as shown in Fig. 178, and as close in to the crank webs as possible, since there can be no reason for keeping them any further apart, and thus necessitating larger castings and more bending in the shaft. During the upward-stroke of the piston the crank shaft is pulled against the top brass, and, therefore, produces a tensile stress in the holding-down studs, and a bending effect upon the cap. The cap is, therefore, designed as a beam, and the studs must be of sufficient area at the bottom of the threads, so that the tensile stress per square inch due to the piston pressure shall not exceed the usual value (see § 15, Table II.)

In the horizontal engine ot Ex. B the crank shaft is supported by separate pedestals of the design of Fig. 158, which are bolted to the engine bed in the position shown in Fig. 179. In considering the distance between the two pedestals, it should be remembered that the greater the distance, the greater the bending effect upon the shaft, due to the pressure on the crank pin; and the wider the bed plate. Altogether the design would probably be more economical if the flywheel pedestal were moved nearer the eccentrics than as shown.

The design of the crank-shaft bearings and the fixing of the exact length of the shaft is best undertaken when designing the engine framing, and will, therefore, be left till then.

EXAMPLES.

EX. A 7.—Make working dimensioned drawings, half full size, of a steel crank shaft for the vertical engine of Fig. 178. Crank pin size to be found from Ex. A 5. Crank webs $1\frac{3}{16}''$ thick, increasing in width from $3''$ to $7\frac{3}{4}''$; length, $9\frac{5}{8}''$, thus forming counterbalance weights. Collars next webs, $\frac{5}{8}''$ thick, $4\frac{3}{8}''$ diameter; shaft diameter, $2\frac{7}{8}''$; journals, $2\frac{7}{8}''$ diameter; pressure per square inch = 150 lbs.; coupling at end, $\frac{7}{8}''$ thick, $7''$ diameter; face of coupling, $10''$ from centre of crank pin. Shaft turned down for eccentric sheave, diameter to be found from Ex A 6.

(Calculate length of journals as already shown.)

EX. B 7.—Make working dimensioned drawings, half full size, of a steel crank shaft for horizontal engine of Ex. B, and as in Fig. 196. Crank pin size from Ex. B 5. Crank webs, $4\frac{1}{4}''$ wide, $2\frac{3}{4}''$ thick; diameter of shaft, $3\frac{3}{4}''$; journals, $3\frac{1}{2}''$ diameter; pressure per square inch = 156 lbs.; weight of flywheel, 8 cwt. Loose counterbalance weights can be fitted, for proportions of which see § 217.

(The webs need not extend in length beyond crank pin and shaft a greater distance than their width extends beyond the diameter. Note increased length of flywheel journal.)

It is unnecessary to show the whole length of the crank shaft, or the exact distance between the journals in either example, hence it will be convenient to draw it broken off in parts, and then the lengths between centres of journals and total length can be left until after designing the engine framing and bed plate.

SECTION XXXI.

JET CONDENSERS AND AIR PUMPS.*

(205) **Condensers.**—Steam-engine condensers are of two kinds—the first and most common are called jet condensers, owing to condensation being effected by the exhaust steam meeting a jet of cold water and mixing with it; the second are known as surface condensers, in consequence of the steam being condensed by coming in contact with a cold surface, and without mixing with the condensing water, the steam passing inside a large number of small tubes and the water passing outside them. Surface condensers are used for engines of large power, especially when the water supply is too dirty to be used as feed for the boilers, but jet condensers are most common for the class of engines we are considering.

(206) **Air Pumps.**—Condensers are fitted with a pump worked from the engine which removes the condensed steam, and in a jet condenser, the condensing or injection water also, from the mixing chamber to a part of the condenser called the hot well, from which it passes to the boilers as "feed." But in doing this the pump also produces a partial vacuum in the space between the mixing chamber and the engine cylinder, by exhausting the

* Students working the examples for the vertical engine of Ex. A should pass over this section to Section xxxii.

air and steam from behind the piston, and is known in consequence as the "*air pump.*" Air pumps are of different forms, but we shall only deal with the type as fitted to the jet condenser of Ex. B, the construction of which we shall describe.

(207) **Jet Condensers.**—In ordinary jet condensers the mixing chamber, air pump, and hot well are contained within the one casting, which is usually of cast iron. In Figs. 197*a*, *b*, *c*, are shown three views of the jet condenser as fitted to the horizontal engine of Ex. B. Fig. 197*a* is a front section through the centre, Fig. 197*b* an end section, and Fig. 197*c* the plan. The condenser casing consists of an oblong box, containing the mixing chamber, M C, the pump chamber, P C, and the hot well, H W. The air pump, A P, is a simple cast-iron plunger, cast hollow, and attached at the front end by means of a cottered joint to the back end of the piston-rod. The exhaust steam enters from the cylinder at the opening, E S I, in the top of the condenser, and meets a jet of cold water which enters from the side through the opening marked C W I, and is directed upwards immediately under the steam inlet by a pipe not shown. The steam is condensed and mixes with the water in the chamber, M C, falling around the barrel of the air pump, into the lower part next the suction valves, S V. As the plunger moves outwards these valves open, and the water passes into the pump chamber, P C. On the return stroke of the plunger as it moves inwards, it forces a certain quantity of the water through the delivery valve, D V, into the hot well, H W, the same process being repeated every revolution of the engine. The air pump is, therefore, single acting—that is, it only discharges each revolution, not each stroke. The hot well is connected to a discharge or overflow pipe at the opening marked H W O, or the water may pass down the sloping division plate, P, into the triangular-shaped passage P, and along to the other side of the condenser from which a pipe marked F P leads to the engine feed pump. The pump barrel, P B, forms the stuffing box for the plunger, the back cover, B C, being for access to the suction valves, and the hot well cover, H W C, for access to the delivery valve.

(208) **Suction and Delivery Valves.**—These valves are of indiarubber, covering gratings of the design shown in Fig. 169, and are fitted together so that a turned projection on the seating fits into holes in the division plates between the chambers. The suction valves are then held in position by the set screws, S S, which pass through the crossbar, C B, shown in the end view, fitting in the lugs, L L, cast on the inside of the condenser. The delivery valve is fixed by the vertical stay, D V S, the upper end of which fits into a groove in the valve seating, while the lower

Fig. 197a.

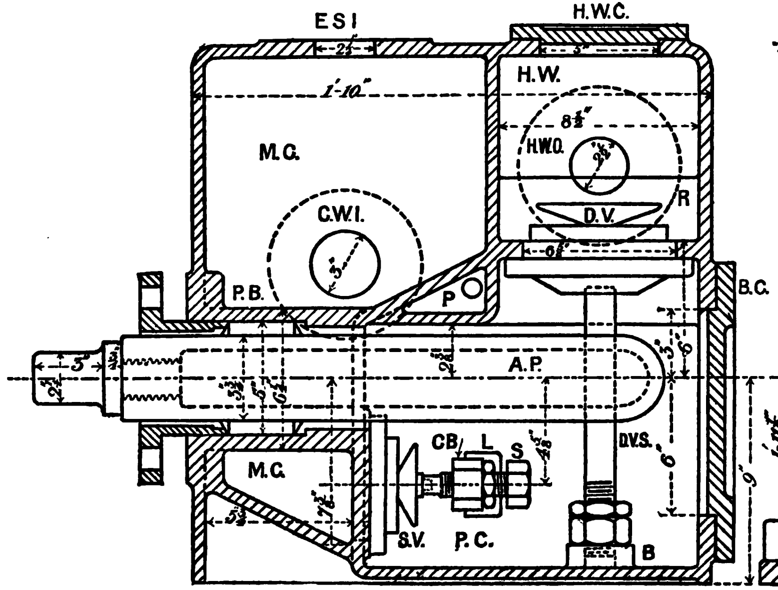


Fig. 197b.

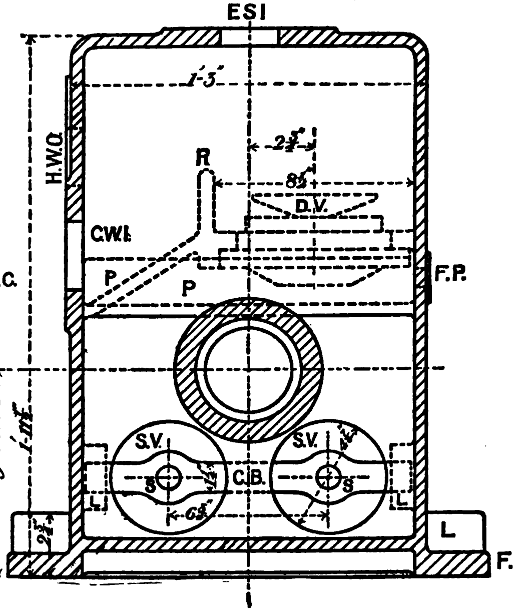
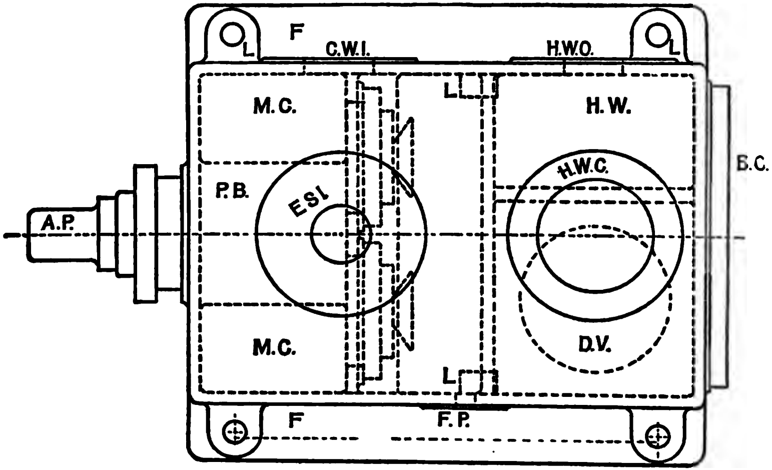


Fig. 197c.



Figs. 197*a*, *b*, *c*.—Index to Parts.

Sectional Front and End Elevations, and Plan of Jet Condenser.

M.C. for	Mixing chamber.	F.P. for	Feed pipe to feed pump for boiler.
P.C. ,,	Pump chamber.	H.W.O. ,,	Hot well overflow.
H.W. ,,	Hot well.	P.B. ,,	Air pump barrel.
H.W.C. ,,	Hot well cover.	B.C. ,,	Back cover.
A.P. ,,	Air pump.	C.B. ,,	Crossbar for fixing suction valves.
E.S.I. ,,	Exhaust steam inlet.	S.S. ,,	Screws in crossbar for suction valves.
C.W.I. ,,	Condensing water inlet	D.V.S. ,,	Delivery valve stay.
S.V. ,,	Suction valves.	F.F. ,,	Flanges for bolting to engine bed-plate.
D.V. ,,	Delivery valve.	L.L. ,,	Lugs or bosses for holding down bolts.
P. ,,	Passage from hot well to feed pipe.		

end is screwed, and rests in the circular boss, B, cast on the condenser bottom, adjustment being effected by the nuts shown. Notice that the indiarubbers of the suction valves are placed inside the pump chamber, while that of the delivery valve is inside the hot well, so that water cannot return from the pump chamber into the mixing chamber, nor from the hot well into the pump chamber. The condenser is provided with flanges along the two sides, lugs or bosses being provided for the holding-down bolts, as shown.

(209) **Injection Spray Pipe.**—The pipe which directs the condensing water upwards to meet the exhaust steam is made of cast iron, is bolted by a flange to the outside of the inlet, C W I, and is turned upwards immediately under the centre of the exhaust steam inlet, and to within $5\frac{1}{2}''$ of it, the end being closed, enlarged in diameter, and pierced by a number of slits or holes so that the water meets the incoming steam in the form of a spray.

(210) **Arrangement of Details.**—Other points to be noticed about the design of such a condenser are—that the exhaust steam inlet should be at the highest point, so that no water may possibly find its way along the exhaust pipe to the cylinder; that the suction valves should be at the lowest point, so that full advantage may be taken of any head of water in the mixing chamber, to assist the working of the suction valves; that the delivery valve should not be too high above the pump; and that the valves, when of the indiarubber pattern, must be arranged so that, when the engine is not working, they remain covered with water, otherwise the rubber would dry and crack. This is effected in the condenser shown by having the suction valves below the plunger, so that when the water level in the pump chamber sinks to below the plunger, no more water will be displaced, and by casting the rib, R, across the hot well, a little higher than the delivery valve, thus forming a kind of trough, which always remains full of water. The hot-well overflow, H W O, must also be above the top of the valve. The object of inclining the bottom of the mixing chamber is to better direct the water against the suction valves.

(211) **Size of Condenser and Air Pump.**—A small condenser is liable to become flooded, and a large condenser simply means so much more space to produce a vacuum in. In the condenser illustrated, the mixing and pump chambers have each a volume of approximately three times the cylinder volume, and the hot well a volume of about 1·3 times. In commencing to design a condenser of this pattern, we first of all recognise that the air pump must be large enough to get rid of the condensing

water when having a speed equal to the piston speed, and that the valves must be large enough to permit the water to pass through them without its velocity being great—the maximum velocity of ordinary practice being 300 feet per minute. But as the height of the air pump centre above the bed plate has already been decided, and also its stroke, we see from the end view, Fig. 197*b*, that we could not get larger suction valves in without increasing the width of the condenser, nor must we have a less length of pump chamber than the air pump-stroke, as seen in Fig. 197*a*, if we are to let the pump displace a quantity of water equal to its own volume, nor, again, can we get in a sufficiently large delivery valve with a less width of hot well. It is such considerations as these which decide the sizes of this particular condenser.

In order to find the air pump diameter knowing its stroke, we must know the volume of condensing water required per minute, and then obtain such a diameter, so that volume of pump × revolutions per minute = volume of condensing water per minute; the volume of steam when condensed being so comparatively small that it may be neglected. The amount of condensing water required is found as follows:—Find the volume of steam used per minute at greatest speed = volume of cylinder × revolutions per minute. Reduce this volume to weight in lbs., by finding from tables of the Properties of Steam the volume of 1 lb. at the maximum terminal pressure. This terminal pressure must be calculated knowing the greatest cut-off, and the highest initial pressure, it being sufficiently near to assume that "*pressure varies inversely with volume.*" Find also from the tables the total heat in 1 lb. of steam at the exhaust pressure from 0° F, which we will call T units. We then require to know the temperature of the condensing water, and the temperature of the hot well. The former of these is usually taken as 60° F. $= t°$, and the latter as $120° = t_1°$. Then if w = lbs. of steam used per minute, W = lbs. of condensing water required per minute, we know that—

$$\begin{aligned} \text{Heat gained by water} &= \text{Heat lost by steam} \\ W\,(t_1° - t°) &= w\,(T - t_1°) \end{aligned}$$

from which we can find W, knowing w, T, $t_1°$, and t. Then express W in cubic inches, knowing that 1 cubic foot of water weighs 62·5 lbs. Let this volume = V cubic inches. We must now make an allowance for the fact that the pump does not displace an amount of water equal to its own volume per revolution, owing to what is called "slip." The usual rule for single-acting plunger pumps is to allow a pump efficiency of 0·5, so that we have—

$$0{\cdot}5\ \{\text{volume of pump} \times \text{revolutions per minute}\} = V;$$

that is $$0{\cdot}5\left\{d_p^2 \times \frac{\pi}{4} \times \text{stroke} \times \text{revs. per minute}\right\} = V,$$

from which we can find the diameter d_p; also to find area of suction or delivery valves we have—

$$\therefore \text{Area of value in sq. ins.} \times \text{velocity of water in inches per minute} = V,$$

from which we can find the valve diameter knowing the area and the velocity of the water. This area is, of course, the area through the grating (see § 139).

The diameter of the inlet for the condensing water depends upon the height from which the water flows, and, therefore, upon its velocity through the pipe. But both the inlet and outlet should be designed so that the velocity of water passing through shall not be greater than 200 feet per minute. Hence, if d = diameter of pipe, then—

$$d^2 \times \frac{\pi}{4} \times \text{Velocity in inches per minute} = V.$$

If the student will work out these sizes for the engine of Ex. B, taking pump stroke 12″, revolutions per minute 150, initial pressure of steam 98 lbs. per square inch absolute, cut-off $\frac{5}{8}$″, condensing water 60° F., hot well water 120° F., velocity through valves 300 feet per minute, he will find that the results agree with the actual sizes.

EXAMPLE.

EX. B 8. Make working dimensioned drawings, half size, of the jet condenser for the horizontal engine of Ex. B, having the sizes given, and as in Fig. 197. Outside measurements 1′ 10″ long, 1′ 3″ wide, 1′ 11$\frac{1}{2}$″ high, thickness of metal $\frac{1}{2}$″. Hot well 8$\frac{1}{2}$″ wide, 8″ deep. Suction valve openings 4$\frac{7}{8}$″ diameter, centres 6$\frac{3}{4}$″ apart, and 4$\frac{5}{8}$″ below air pump centre. Delivery valve opening 6$\frac{5}{8}$″ diameter, centre 2$\frac{3}{4}$″ to one side of centre line of plunger. Faces for suction valves 6$\frac{3}{4}$″ from outside of condenser front. Plunger 3$\frac{3}{4}$″ diameter, thickness $\frac{5}{8}$″, total length 1′ 11,″ diameter outside barrel 6$\frac{1}{4}$″, length from face to face 7″. Diameter of cast-iron spray pipe inside 1$\frac{1}{2}$″. Outlet from hot well 2$\frac{1}{2}$″ diameter; outlet to feed pump $\frac{3}{4}$″ diameter. Steam inlet 2$\frac{1}{2}$″ diameter; water inlet 3″ diameter. Crossbar for suction valve 1$\frac{1}{4}$″ square; rod for delivery valve 1$\frac{1}{4}$″ diameter. Height of centre line of pump above base 9″; size of exhaust steam inlet 2$\frac{1}{2}$″. Holding-down bolts 1″ diameter. Faces for machining to be provided at covers, stuffing box face, and for suction valve

and pipe flanges. Make a detail drawing in another part of the paper of one of the suction valves. Rubber 4¼″ diameter, ½″ thick; guard 3¾″ diameter (see § 139).

(First draw in centre lines and the outside outline, then draw in plunger and barrel; next show thickness of metal and division plates; then draw in suction valves, delivery valves (in outline only), and inlet and outlets. Make the covers on the pump chamber and hot well as large as possible, and show the bolts. Draw in stuffing box for plunger (proportions as in Section xxiv.) and flanges for bolting down; width to allow for use of spanner on nuts. Finish details. Notice that the lugs for crossbar and upright rod must be open at one side, and that sides and ends of condenser come below bottom to give parts for machining. Work all views together.)

SECTION XXXII.

ENGINE FRAMES AND BED PLATES—GENERAL ARRANGEMENT DRAWINGS.

(212) **Engine Framing.**—Having completed the drawings of the cylinder and its connected parts, it now remains to design the engine framing or bed plate. We start by knowing that the framing is required to support the different parts in as convenient a manner as possible, generally in as little space as is consistent with the requirements of a rigid base, and that the distribution of metal must be such that when the engine is working at its maximum speed and power, there shall be no lack of stiffness and strength. It is probable that engine framings are designed from proportions and forms suggested by the results of successful practical experience, rather than by theoretical considerations of the special stresses acting in certain parts, which stresses are in many cases almost beyond determination, so that we shall only approach it from the aspect indicated above.

As a preliminary step, we must decide the length from the centre line of the cylinder to the crank-shaft centre. To do this we make a rough sketch of the parts when the piston is at the back end of the stroke, and the crank on one dead centre as in Fig. 178. We then see that the distance is equal to—

{Length of crank (or ½ piston-stroke) + length of connecting-rod + distance from centre of crosshead to end of crosshead nearest cylinder + clearance between end of crosshead and gland studs + distance from end of gland studs to centre of cylinder.}

All these separate sizes can be obtained from drawings previously made, except the clearance between crosshead and gland, and this we see must be determined by knowing how near the end of the slipper or guide block, if it extends beyond the crosshead may come to the cylinder, and also that until this distance is settled we cannot rightly fix the length of the piston-rod. In the vertical engine of Ex. A (Fig. 178) we see that the slipper cannot conveniently approach nearer the cylinder than the distance shown, owing to the position of the guides below the top flange of the standard, and also that in Ex. B (Fig. 179) the least distance is decided by the length required for the lugs on the cylinder end to which the guide bars are bolted, although in this case it will be found that a clearance between the cross-head and gland studs gives ample clearance for the guide block.

It will be necessary then for the vertical engine of Ex. A to design and draw out the framing before the limiting position of the crosshead can be fixed. From the drawing of Fig. 178 it will be seen that the bottom framing for the crank-shaft bearings, the back standard, and the top flange for the cylinder form one casting, the shape of which will be clearly seen from the views given. The bottom box section framing is raised at the sides to receive the shaft bearings C B, and to provide bosses for the front columns, it is also recessed under the crank as shown by the dotted line *e, f, g, h*, Fig. 178, B, to provide clearance for the crank and its attached parts. The back standard is known as the A pattern, rising from the bottom frame as two legs, which join at the line *m n*, and then continue as one piece up to the top flange, the shape of which is seen in the plan and which is also provided with bosses for the front columns. A section plan across the line *a b* is shown in Fig. 178, D, which will make the construction of that part clear, and which shows how the flat face *p* is left to form a surface for the slipper guide. In determining the height to the top of the flange from the shaft centre, the connecting-rod and crosshead with the slipper should be drawn in outline, showing merely limits of lengths, and then the clearance of about $4\frac{3}{4}''$ allowed between the slipper and the top of the flange. Care should be taken that this gives sufficient clearance between the extreme top parts on the crosshead and the cylinder gland.

EXAMPLES.

EX. A 8.—Make working dimensioned drawings of the casting for the framing and standard of the vertical engine of Ex. A, as in Fig. 178, showing at least three views. Scale, $3'' = 1'$.

Height from base to centre of crank-shaft bearings, $8\frac{3}{4}''$; thickness of metal, $\frac{1}{2}''$; of bottom and top flange, $1\frac{1}{4}''$; around bearing $1\frac{1}{8}''$; around front columns at bottom, $1''$; bosses at top, $3''$ diameter, $3\frac{1}{2}''$ long. Columns, $1\frac{3}{8}''$ diameter. Holding-down bolts, $\frac{7}{8}''$ diameter. Other sizes as in Fig. 178.

(Obtain the distance between the bearings centres from crank-shaft drawing, Ex. A 7, knowing crank pin, webs, and length of journals. Draw in circle for journal in Fig. 178, A, allow thickness for brasses (see § 115), and thus obtain height to top of crank-bearing frame. The width of the frame should be less than total width across the brasses, by an amount equal to (twice the flange thickness on the brasses + allowance for facing for flanges), as these faces should be machined without touching the surrounding metal. The distance from the centre line through the piston-rod to the face for the slipper guide is found from the crosshead drawing, and also the length of face required, thus giving the position for the joining of the two legs at the line $m\,n$ (least length of face = length of slipper + stroke, less allowance for slipper to overlap guide plate at ends of stroke). See Fig. 187 for sizes of slipper-guide plates. The flange diameter at the top need only be about $\frac{1}{4}''$ larger than the flange on the cylinder which is bolted to it. The front columns, as seen in Fig. 178, B, are fixed so that the distance of their centres apart is equal to the distance between the centres of the bearings. The top ends are turned smaller in diameter and screwed into the bosses on the flange, and the bottom ends are fixed by round pins, about $\frac{5}{8}''$ diameter, slightly taper, driven through rod and boss. Flange width at base to suit holding-down bolts. The least distance from the engine centre to the legs of the A frame, where they leave the bottom framing, is settled by the length of the bearing cap, which should, therefore, be drawn in position as a guide.)

EX. B 9.—Make working dimensioned drawings, three views, of the bed plate for the horizontal engine of Ex. B, as in Fig. 179. Scale, $3'' = 1'$. Total depth $4\frac{1}{2}''$, $1''$ less at condenser end to allow for extra depth of condenser below engine centre. Flange, $1\frac{1}{4}''$ thick; thickness of metal, $\frac{5}{8}''$; seven cross ribs, $\frac{5}{8}''$ thick. Holding-down bolts, 12—$1\frac{1}{4}''$ diameter, passing through each corner of plate, and through ribs which are enlarged by bosses where they join sides of plate, to allow of bolt holes. Width over all at condenser part, $2'$; at cylinder part and for rest of length, $2'\ 7\frac{3}{4}''$. Distance from shaft centre to end $16''$. Bed plate to be recessed under crank for clearance, the recess to continue for a less width, and decreasing in depth, forming a groove $4''$ wide from the crank recess up to the cylinder face, and under the glands of the cylinder and steam chest, to form a dish for the oil. Facings for planing $\frac{1}{4}''$ thick to be provided under condenser, cylinder, guide standard, valve-rod guide, and pedestals for shaft. Distance between centre of cylinder and centre of condenser, $3'\ 10''$. Other sizes as in Fig. 179.

(The design of this bed plate should present no difficulty, but it will be necessary to refer to previous drawings. The best way of proceeding will

be to draw the plan of the plate, first draw engine centre line and centre lines of condenser and cylinder, then show the outline of the condenser and cylinder flanges which are to be fitted to the bed plate, and draw the plan of the facings for these, allowing about ¼" beyond all round. Next show the outline of the feet of the guide standard, obtaining sizes and distance in front of cylinder centre from drawings of Figs. 179 and 190, and arrange facings in same way. It will then be better to draw the outline of the front cylinder cover, showing the lugs for the guide bars, and the guide block and crosshead in their nearest position to the cover. Then mark off along the centre line from the crosshead centre, a distance equal to length of connecting-rod + length of crank (Ex. B 5 and 7), and this will give position of centre of crank shaft. Now mark off along the shaft centre line, the limit of the crank web in one direction, and the expansion eccentric in the other (see Figs. 181 and 196), then fix the centres for the shaft pedestals (Fig. 158), which may conveniently be as near to edge as ½". Mark out the facing for the pedestal bases as before, and also for guide bracket for valve-rods. Next draw the plan of the recess for the crank, and for catching the oil, the crank recess should extend to cover all positions of the connecting-rod end. The cross ribs may equally divide the length of the bed plate, the bosses for the bolts being about 3½" within the edge; care must be taken that they do not foul with other bolts for the cylinder, condenser, or pedestals. Notice that the flange need not project for more than ½". Then draw a front elevation and an end section, and dimension.

(213) **Length of Piston-Rod, Valve-Rod, and Crank Shaft.** —From what has been already said we are evidently now in a position to exactly settle the length of these parts. For the piston-rod of Ex. A, its length from the crosshead centre to under side of piston will be equal to—

(Distance from centre of crosshead to top of flange on standard when at back end of stroke + distance from under side of flanges on cylinder to inside of front end, + clearance + length of stroke of piston),

and for the rod of Ex. B, it will be equal to—

(Distance from centre of crosshead to inside of front cover, + clearance + stroke);

while to obtain the total length of the piston-rod since it passes through the back cover to join the air pump, we must add a length of—

(Distance from centre of cylinder to centre of condenser + length from centre of condenser to inside back end − clearance for plunger from end − total length of plunger + length of piston-rod in plunger end for cottered joint − half piston stroke.)

Allowance must be made for the length of rod required to fix to piston and crosshead.

To find the length of the valve-rod we must decide the position of the guide bracket and the length of the guide on the valve-rod (the enlarged part of valve-rod, see Figs. 184, 185, 186). Then we can find the distance from the centre of the valve at mid-

stroke to the outer edge of the guide bracket, and add half the valve travel and the distance from centre of pin in forked end of valve-rod to edge of guide in nearest position to cylinder. The length of the eccentric-rods is best found from a drawing where the piston is shown at one end of the stroke and the eccentrics in their corresponding position upon the engine shaft (see Fig. 194).

Having determined the positions of the crank-shaft bearings, we are able to dimension the length of the crank shaft. It is purely a matter of convenience how far the shaft overlaps each end, except when a flywheel or driving wheel is fitted, and then, at least, a minimum length equal to the wheel boss is necessary.

(214) **General Arrangement Drawing.**—A general arrangement drawing should show the engine complete, with the different parts in their correct relative positions. It should be something of the character of Figs. 178 and 179, only that more parts should be shown, and at least three views, two elevations and a plan, be drawn. In a finished drawing, parts may or may not be in section, usually not, and such details as bolts and nuts, lubricators, steam valves and pipes, and drain cocks may often usefully be included. The student should find no difficulty in making such a general arrangement drawing after having worked the previous questions, as the chief difficulties have been explained. If the cylinder is shown in section, it is usual to draw the piston at one stroke end, and the valve at mid-stroke, these being the easiest positions. There are other details, such as governors, feed pumps, which may be attached to the engine, but they may be so separate as not to affect any of the other work, and as space is limited they are omitted.

EXAMPLES.

EX. A 9. Make a general arrangement drawing, three views, of the vertical engine of Ex. A (Fig. 178). Scale $1\frac{1}{2}'' = 1'$.

EX. B 10. Make a general arrangement drawing, three views, of the horizontal condensing engine of Ex. B (Fig. 179). Scale $1\frac{1}{2}'' = 1'$. Crank shaft pedestals as in Fig. 158.

SECTION XXXIII.

DESIGN OF MACHINE PARTS OF GIVEN WEIGHT.

(215) It is very desirable that the engineering draughtsman should be able to design parts which, when finished, shall not exceed a certain weight. The necessity for this frequently arises with such details as governors, counterbalance weights, and the flywheels for steam, gas, and oil engines, especially the last two, which depend so much upon the flywheels for their uniform running. Cases often occur, also, where probably a whole engine has to be produced under restrictions and penalties as to its total weight—such, for example, as the engines for war ships, or traction engines for service over light bridges in foreign countries—and the satisfactory fulfilment of the order may rest upon this question alone. That such examples present considerable difficulty is apparent, for the draughtsman has not only to fulfil all the usual conditions of strength and proportions, but he has to exercise a large amount of ingenuity to so distribute his necessary material in the most economical way as to secure the result of least weight.

This section is included in order to give the student some examples of this kind, and it was for this purpose that the tables of pp. 162-165 were extended to include the weights of unit volumes of the ordinary materials of construction, and of the standard sizes of bolts and nuts. Of course, the results can only be approximately assured, owing to the difficulty of making a correct allowance for the fillets of corners, and similar parts, but after some practice, the designer acquires the knowledge of many ways in which these are very nearly accurately allowed for without undue complication. The examples chosen for working will not require anything beyond an elementary knowledge of the mensuration of surfaces and solids.

In working the examples, the student should make sketches of the part, filling in the sizes of the parts whose dimensions are fixed by other considerations than the weight. He should then try to complete the design roughly to fulfil the conditions of weight also, before beginning the actual working drawing. Then in making the drawing, he will see that there are certain sizes which must not be altered, and certain others—such as overlap of flanges, thickness and diameter of bosses—which can be altered without loss of strength.

EXAMPLE.

EX. 1.—Design a simple shaft pedestal, as in Fig. 154, Ex. 1, p. 227, so that its weight, when finished, shall be between 3 and 4 lbs.

(Here the only fixed size is the length and diameter of the brasses, and we can easily save weight in reducing the height from the base to the centre, since this is not stated, and in the base length. First find what weight the ordinary proportions will give, and then, if too heavy, cut away the least important parts.)

(216) **Flywheels.**—In a steam engine having a crank and connecting-rod, the turning effort on the crank pin varies considerably during the stroke, owing to the different angle between the crank and rod, the changing pressure upon the piston, and the change in the velocity, and, therefore, in the momentum of the moving parts. At the two dead centres the turning effort is evidently *nil*. Hence, if the resistance offered to the engine is uniform, it follows that at some parts of the stroke the power of the engine is in excess of the resistance, while at other parts it is less, and this must mean that the velocity of the crank pin will slightly change. But if a heavy flywheel is fitted to the shaft, it decreases the amount of this change of velocity by absorbing energy when the work done on the crank is greater than required for the resistance, and by giving it out again when it is less. It does this by slightly changing its velocity, and in virtue of the fact that the energy of a rotating body depends upon its mass and its velocity; hence, since the mass of the wheel is constant, its velocity changes in order that the amount of its kinetic energy may change. But if the amount of the variation in the crank effort is found, as it can be by well-known methods, it is possible to design a flywheel of a certain size and weight, so that its actual variation from a uniform velocity may be of any desired value. This variation for well-balanced engines does not exceed 1 to 2 per cent.

We shall not deal with the methods by which these results are arrived at, but simply consider the question of designing the wheel when they have been decided. In doing this, we notice that the wheel is most economically designed when the greatest mass is at the greatest distance from the centre of the shaft, and this means that the mass should be concentrated as far as possible at the wheel rim. Purely practical conditions decide the diameter of the wheel in individual cases, and we shall assume the diameter is given as well as the sizes of the boss and spokes, and the greatest width of the rim.

EXAMPLES.

EX. 2.—Make working drawings to a scale of $1\frac{1}{2}'' = 1'$, of a cast-iron flywheel for use with the engine of Ex. B, Fig. 179. Shaft, $3\frac{3}{4}''$ diameter; boss, $7''$ diameter, $7\frac{1}{2}''$ long; six spokes, elliptical section, $3\frac{3}{4}'' \times 2''$ at boss, tapering to $3'' \times 1\frac{11}{16}''$ at rim; outside diameter of wheel 5′ 6″, width 8″, rim to have inside flanges $1\frac{3}{4}''$ wide from inside of rim and $\frac{1}{2}''$ thick. Total weight of wheel 8 cwts.

(In drawing the wheel it will be found that the spokes meet before reaching the boss. They must be made to curve into what may be called a second boss, having a width equal to the spokes at that part, and a diameter so that a curve of about 2″ radius will join two spokes and the boss. After drawing as much as is given in the question, the weight of the parts drawn must be found, and the difference between the wheel weight and their weight will be the weight of the wheel rim, which must then be designed of a given thickness, knowing its width, so as to weigh that amount. It will be noticed that until the thickness of the rim is known, the real length of the spokes is not really known from which to find their weight, but it will be sufficiently near to assume that they reach to the outside of the wheel.

The drawing of a flywheel should show a view looking on all the spokes, and a second view looking on the wheel rim, half of which should be a section through the rim and boss.)

EX. 3. Make working drawings to a scale of $1\frac{1}{2}'' = 1'$ of a fly for a gas engine. Shaft $4\frac{5}{8}''$ diameter. Boss $9\frac{1}{4}''$ diameter, $8\frac{3}{4}''$ long. Six curved spokes of elliptical section $4\frac{5}{8}'' \times 2\frac{3}{4}''$ at boss, $3\frac{3}{8}'' \times 2\frac{1}{4}''$ at rim; outside diameter of wheel 5′ 10″, width of rim 8″, rim of T section central web about 3″ thick and $5\frac{1}{2}''$ deep. Total weight 22 cwts.

(217) **Counterbalance Weights.**—Quick speed engines run much more smoothly when weights are attached to the crank webs on the opposite side to the crank pin, to balance the weight of the parts acting at the pin. It is usual to take half the weight of the connecting-rod as acting at the crank pin, and the other half as acting at the crosshead. The weight of the crank pin, the crank webs beyond the shaft, and half the connecting-rod, are then taken as rotating about the shaft centre at a radius equal to the crank length, while the weight of the piston-rod, crosshead, and the other half of the connecting-rod are supposed to act together along the centre line of the engine. The former affect the balance both in the line of the stroke and also at right angles to it, but the latter only influence the balance in the direction in which they move—that is, in the line of the stroke.

In vertical engines a want of balance at right angles to the line of the stroke is found to be most injurious, and in a horizontal engine a want of balance along the line of stroke. Both cannot be balanced at the same time, and hence for vertical engines we consider the former and for horizontal engines the latter. If the total weight concentrated at the crank pin = W, the total weight concentrated at the piston = W_1, the counterbalance weight = W_2, the crank radius = r, and the distance of the centre of gravity of the counterbalance weight from the shaft centre = d, then to produce balance at right angles to the line of the stroke—

$$W_2 \times d = W \times r.$$

Hence for vertical engines we use this rule to find $W_2 \times d$, so that their product is equal to W r. For horizontal engines, Prof. Unwin states that a usual rule is—

$$W_2 \times d = \tfrac{2}{3} (W + W_1)\, r.$$

These rules are only approximate, for it is not absolutely correct to suppose the connecting-rod divided as in the above, nor ought we to take the centre of gravity of the counterbalance weight instead of more accurately its radius of gyration; but such a method is always adopted in work of this kind, the really accurate methods not giving results which differ sufficiently to make the difference at all practically appreciable.

That this latter statement is true will be recognised when we point out that the volume of the counterbalance weight required in order to fulfil the above conditions is often too great for practical adoption, and that the designer has to be satisfied with only balancing a part of the unbalanced masses. This we see when we know that the width of a counterbalance weight should not much exceed the thickness of the crank web, otherwise it will foul the connecting-rod end, nor should its distance from the shaft centre be greater than the most distant part of the connecting-rod at the crank, otherwise it requires an increased clearance from the engine bed-plate.

EXAMPLES.

EX. 4. Design counterbalance weights for the crank shaft of the vertical engine, Ex. A (see Fig. 178 and Ex. A 7, Section xxx.), to form a continuation of the crank webs, and to balance the crank pin, crank webs, and half weight of connecting-rod concentrated at the crank.

EX. 5. Design counterbalance weights for the crank shaft of the horizontal engine, Ex. B (see Fig. 196 and Ex. B 7, Section xxx.),

to form a continuation of the crank webs, and to balance the crank pin and webs only.

(First calculate weight of pin and webs, then find weight required if at a distance equal to the crank radius. See how this weight will distribute in width and thickness, and if badly take a less or greater length for distance of centre of gravity of weight, and distribute again.)

APPENDIX.

MISCELLANEOUS EXAMPLES IN DRAWING AND DESIGN OF MACHINE PARTS, SELECTED FROM EXAMINATION PAPERS OF THE SCIENCE AND ART DEPARTMENT, AND VICTORIA UNIVERSITY.

EX. 1.—A foundation bolt of diameter d'', with a square end, is secured by means of a cotter (see Fig. 131). Find the dimensions of the bolt and cotter, in terms of d, in order that the shearing stress on the cotter may be three-fourths, and the intensity of the bearing pressure on the cotter twice the tensional stress on the bolt. (*S and A. H.*, 1886.)

EX. 2.—A wrought-iron shaft is required to transmit 80 H.P. at 100 revolutions per minute. Draw one of the halves of a cast-iron face-plate coupling for the shaft, and a bolt for connecting the two halves of the coupling together. Scale, $\frac{1}{4}$ full size. (*Vict. Hon. B.Sc.*, 1889.)

EX. 3.—Determine the diameter of the 8 bolts of a flange coupling for a shaft 14″ diam., which is subjected to torsion only, the diameter of the bolt circle being 23″. Draw and dimension the coupling. (*S. and A. H.*, 1887.)

EX. 4.—Determine the necessary depth of the rectangular section of the guide bar of an engine, in order that the maximum stress in the material may not exceed 5 tons per square inch. Total pressure of steam on piston = 25 tons, length of connecting-rod = twice stroke, width of guide bar = 10″. The greatest obliquity of the connecting-rod may be taken to occur when the guide block is at the centre of the span of 4′. (*S. and A. H.*, 1892.)

EX. 5.—Make working drawings of a connecting-rod end of the marine type (Fig. 189). Diameter of rod $2\frac{1}{2}''$, of crank pin 4″. (*Vict. Hon. B. Sc.*, 1890.)

EX. 6.—Design a double chain-riveted (Fig. 142*b*) cover plate joint for boiler plates $\frac{3}{4}''$ thick, and discuss the proportion which the strength of the joint will bear to that of the plates. (*Vict. Hon. B. Sc.*, 1890.)

EX. 7.—A tie rod for a roof is to be designed to stand a tension of 3 tons, with a stress of 4 tons per square inch. It is forked and fastened by a pin to the plate which is 1″ thick. Design and draw the end of the rod—full size. (*Vict. Hon. B. Sc.*, 1892.)

EX. 8.—Draw a stuffing box and gland for the upper cover of a vertical steam cylinder from the following data:—Piston-rod 3″ diameter, cylinder cover $1\frac{1}{4}''$ thick, gland 5″ deep, gland bolts (two) 1″ diameter. The gland to be of cast iron, and both gland and cylinder cover to be bushed with brass. (*Vict. Hon. B. Sc.*, 1892.)

EX. 9.—Draw a cast-iron flange coupling to the following dimensions. Scale, half full size:—

Diameter of shaft 3", diameter of boss of shaft 3⅝", diameter of boss of coupling 7", diameter of bolt circle 10¼", six ⅞" diameter bolts, coupling 9" long, flanges 1½" thick. Show suitable keys for attaching the flanges to the shafts. The bolt heads and nuts need not be sunk into the flanges. (*Vict. U. Ord.*, 1892.)

EX. 10.—Make a full-size drawing of a knuckle joint for connecting two wrought-iron bars of 1½" diameter. (*Vict. U. Ord.*, 1892.)

EX. 11.—A steam engine has a piston of 100 per square inch area, stroke 1½', connecting-rod 3½' long. The steam pressure during admission is 60 lbs. per square inch absolute, and the back pressure 5 lbs. per square inch, cut-off at ½ stroke. Design and draw a crosshead for the engine to work between two parallel guide bars. (*Vict. U. Hon.*, 1891.)

EX. 12.—Draw a cast-iron pulley, with six curved arms, from the following data:—Extreme diameter 40", width of rim 9", thickness of rim at edge ⅜", diameter of shaft 4", thickness of boss 1⅛", length of boss 6", section of arms at boss 2¼" × 1¼". Scale, ¼. (*Vict. U. Ord.*, 1891.)

EX. 13.—Design and draw the crank end of a connecting-rod of the marine type from the following data:—Diameter of steam cylinder 24", initial pressure of steam 100 lbs., diameter of crank pin 5½", length of crank pin 7". Scale, 6" = 1'. (*Vict. U. Hon.*, 1888.)

EX. 14.—Two lengths of a mild steel rod of rectangular section, 7" × 1", are to be connected by means of a riveted butt joint with a cover plate on each side. Design and draw the joint and estimate its efficiency. (*S. & A. H.*, 1893.)

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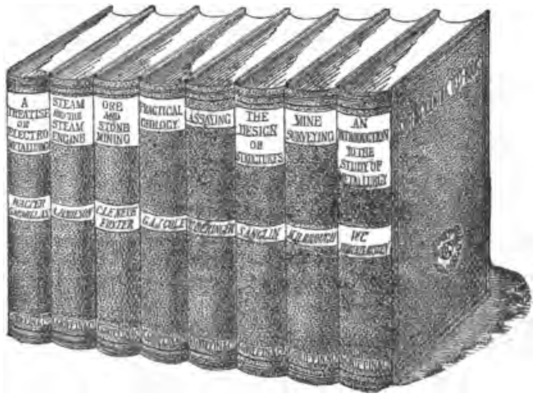

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